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Heat Transfer Solutions for the Phillips Electronics Company

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A THERMAL SOLUTION FOR OVERHEATING LIQUID CRYSTAL DISPLAY COMPONENTS

A PROJECT FOR THE PHILIPS CONSUMER ELECTRONICS COMPANY

Instructor: *Dr. Roger Parsons*

*Mechanical Engineering **479***

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Mr. Raetzer,

On January 18, 1996, the Philips Consumer Electronics Company requested an in-depth study of the heat transfer problem associated with the LCD components within the current prototype of projection television sets. Contained in the following report is an extensive study of various cooling methods available on the market that are specifically designed for electronics applications. In addition to this background information, testing procedures and results are provided for several alternatives that we felt may solve the current heating problem. Then, a final design configuration is described in the last portion of the report based on the preliminary testing results and initial modeling calculations. We appreciate the opportunity to work with the Philips Company and thank you for your cooperation.

Sincerely,

Team #1

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PERSONAL INVOLVEMENT

Before I describe my personal involvement with this project, it would be beneficial if I explained the actual problem dealt with in the design process. This project was initiated by the Philips Consumer Electronics Company. It involved an overheating problem with the three liquid crystal display components of the current production model of the Philips projection television set. In this TV, the overheating associated with the LCD components was due to several factors. The first heat source was that of the internal 200 watt light bulb used to provide the picture. This lamp was the main source of heat and raised the surface temperature of the LCDs above the acceptable 65 degrees Celsius temperature limit. The other sources of heat transfer to the LCDs were several heat generating electronic components mounted directly to the housing that held the liquid crystal displays. The combination of these effects raised the operating temperature of the LCDs to an unacceptable level, thereby inducing operational failure in the form of picture inhomogineity. The purpose of our design project was to formulate a thermal solution for this problem and to devise a temperature measurement scheme in order to prove the effectiveness of the new system.

As the team leader of our design group, my personal involvement with this project included many different roles and responsibilities. First and foremost was that I made all assignments for each team member each week. At the beginning of each week, I talked with my teammates and discussed our short-term goals for that week as well as some long-term considerations that we needed to keep in mind. Then, on each Thursday I would submit a written summary of the assignments I had made to our advisor, Dr. Parsons. The biggest difficulty I faced in this position was that of impartiality. It was difficult at times to make sure that I was being fair to each team member when making certain assignments. I found that sometimes there would be particular tasks that one person had been working on for a few weeks and only he would really have anything to do. As a result, I had to reevaluate tasks and the people working on those tasks to make sure that everyone was doing about the same amount of work. I tried to always ask my team members what they thought about the assignments and to ask for input on what we should be doing. This leads to the second responsibility associated with my position. Even though our advisor would answer any questions we had about a particular problem, it was our job to know what direction we should be taking from the initial background studies to the final design configuration. Consequently, I had to know and pre-determine what we needed to be doing in the classroom at our next meeting. In some cases, this presented a problem because I did not know what we should do next. We met in four hour blocks on Tuesday and Thursday and wasting those precious hours sitting around thinking of what we should do next delayed our design schedule. The last major responsibility that I had was that of putting our three design reports together and making sure that they made coherent sense. This often was a long and arduous process that took hours of meticulous detailing to make the reports look good. I generally am a perfectionist and this made the task that much more time consuming.

If I could go back to January (although I really wouldn't want to) and do certain things over, I think that I would have tried to focus on only a few areas or alternatives instead of trying to use everything that we could find. I felt at times that we were going in ten different directions and that at least eight of them were dead ends. I feel that if we had focused on just two or three cooling methods, our job would have been a lot easier. Instead, we had to sort through enormous amounts of information to find useful ideas. Also, I think that I would have made assignments in a different way. As I mentioned earlier, I usually made the assignments and then asked the guys what they thought about it. Now that I think about it, if I could do it over, I would have first asked them what they thought we should do so that they would not have any preconceptions or preconceived notions about what I was thinking. It is so easy to just say "that sounds good to me" and not really give any more thought to a particular situation or question. In that way, I think that I would have received more useful input from the other team members. The final thing that I would have done differently is the experimental testing. Although I created all the experimental testing procedures that we used in the lab and tested several cooling alternatives, not everyone took part in the testing process due to the incredibly small space that we were given to work in. If I had it to do again, I would have tried to find a larger work area for all of us to use. The room that we were given was so cramped that even the most avid spelunker would have developed claustrophobia.

I think that I learned several things from this experience. Definitely the most important was that of being able to work in a group dynamic. I realized that we were four totally different individuals with different backgrounds working for a common goal. Even though we all got along well the majority of the time, we also had our share of difficult and stressful situations. The most aggravating thing for me was the difficulty we had with everyone meeting outside of class. Everyone's schedules conflicted seemingly every time we needed to get together. This presented a serious problem when the reports were due. However, I think that we worked well together under pressure and were able to pull everything together in the end. Another thing that I learned from this experience was something that I alluded to earlier. This was the fact that, in general, people (engineering seniors in particular) will "go with the flow" and not come up with creative ideas if another (possibly inferior) idea is already present. This was a problem because part of our project assignment was to come up with a "creative" solution to the problem at hand. To battle this tendency, however, we tried to conduct brief brainstorming sessions in order to get the creative juices flowing.

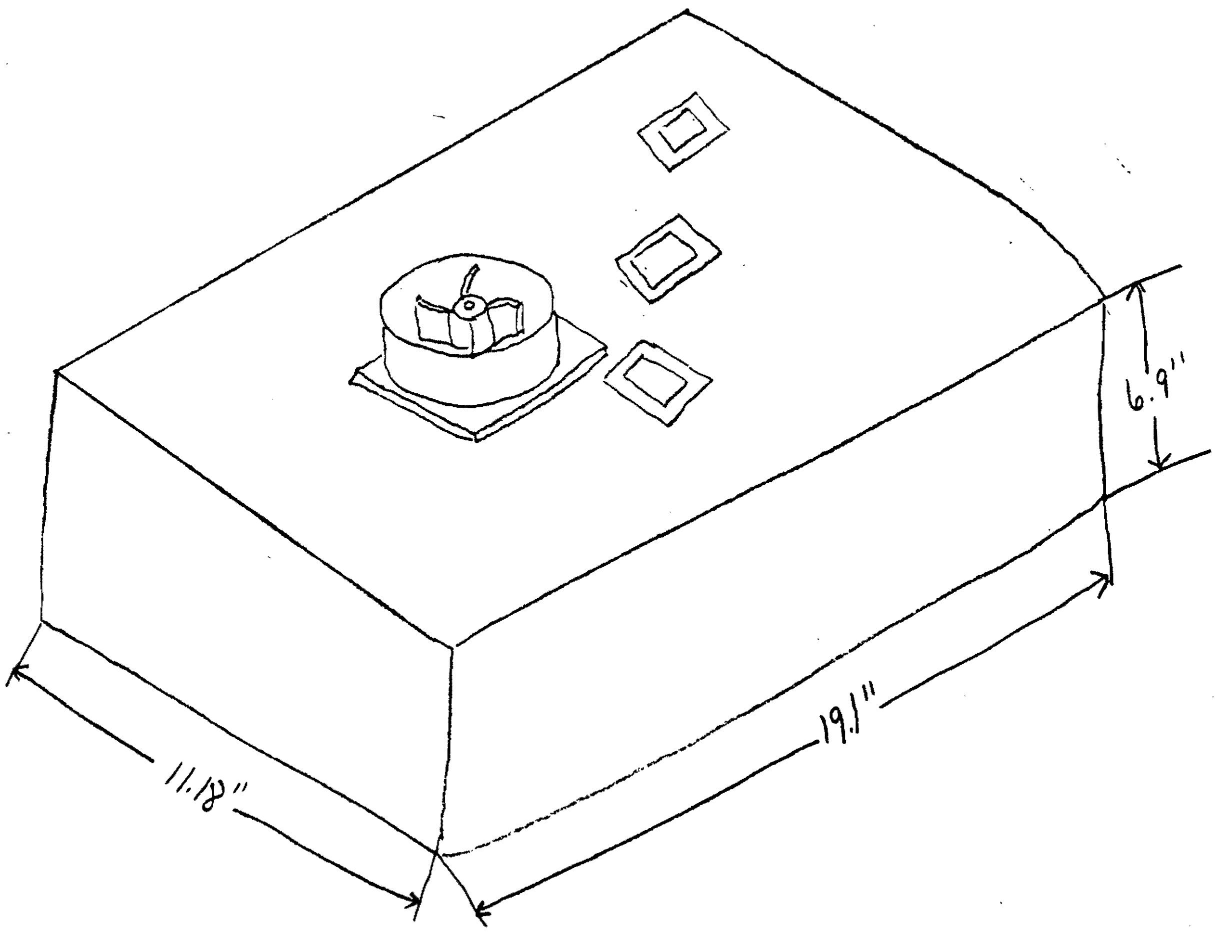
Finally, I would like to talk about the final design solution that we devised for the television set. In the final design, we decided to employ a modified forced convection system using two centrifugal type fans in conjunction with a thermoelectric cooling device. Based on numerous experimental tests, we determined that this solution would be most beneficial on this basis of performance, cost, and reliability. I wish that we would have had more time to actually fabricate the pieces that we designed and described in the final design report. I think that it would have been interesting to actually see the end result of something that we had put our time and effort into for the past three months. Dr. Parsons said that this project is extremely indicative of the kind of problem solving situation that we will be involved with in a professional environment. I am glad we had this opportunity to get a little practice before we actually enter the workplace. In my mind, this has been an eye opening experience and I am sure that it will invariably serve as a positive example of teamwork for me in the future.

PROBLEM STATEMENT

The main purpose of this design project for the Philips Electronics Company is primarily two-fold in nature. The first problem centers on the existing forced convection cooling system. This system is currently unable to cool the three LCD (Liquid Crystal Display) components within the projection TV cabinet as shown in Figure 1. Having been reduced in size by almost 50 % since the last generation, the LCD components still receive the same amount of incident light from the internal lamp and are consequently heated to critically high temperatures. In addition to the incident light, other factors such as heat generating integrated electronic components mounted directly on the LCD and higher ambient air temperatures within the TV cabinet also contribute to unacceptable operating temperatures for the LCDs. In order to function properly, the liquid crystal material inside the LCD housing must remain below 65 degrees Celsius or the unit will malfunction. This point introduces the second problem proposed by the Philips company. In order to experimentally prove the effectiveness of a new cooling system, a temperature measurement method must be conceived so as to measure the steady-state operating temperature of the liquid-crystal material within the LCD itself. This task offers a substantial challenge since conventional methods of temperature measurement cannot be utilized.

In solving this complex problem, a cooling system must be devised to ensure proper functionality of the LCD below 65 degrees Celsius. The design of the improved cooling system must be as non-intrusive as possible. Consequently, anything in direct contact with the LCD components must allow free movement so that the LCD's precise positioning will not be compromised. As a precautionary measure, a temperature measurement system will be developed in order to prove the new system's effectiveness.

Figure 1. Original Forced Convection Cooling System



POSSIBLE COOLING METHODS FOR THE LCD COMPONENT

The cooling methods described below represent a wide spectrum of possibilities that could be utilized in a number of applications. The best techniques are discussed in the body of this report, while several other cooling alternatives are left as optional research material presented in Appendix A.

THERMOELECTRIC COOLING

As one alternative to the heating problem encountered by the Philips Company, thermoelectric cooling was considered as a possible solution to the existence of excessive heat dissipation in the LCDs used in current production. As shown in Figure 2, thermoelectric devices are small solid-state heat pumps that are often used to cool electronic components where space is limited and high reliability is essential.

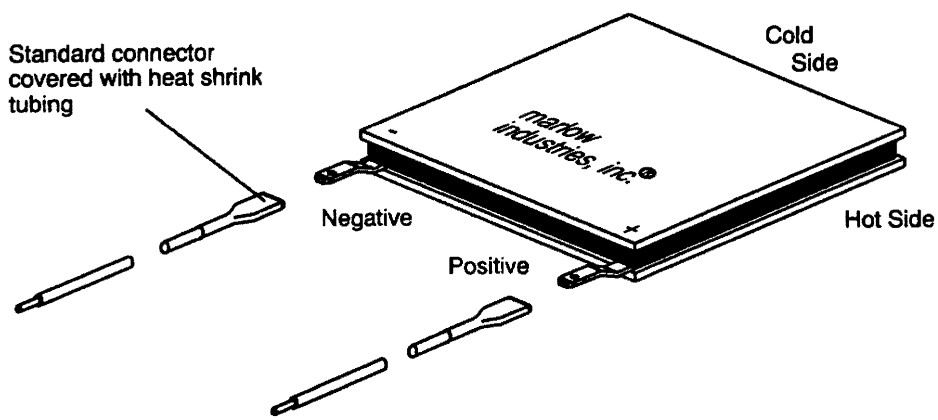


Figure 2. Basic schematic diagram of a typical thermoelectric cooler.

Thermoelectric coolers (TECs) operate on the Peltier principle which states that heat is released or absorbed if an electrical current is passed through a junction of two dissimilar materials. Modern thermoelectric coolers consist of single or multiple stage thermoelectric couples that are electrically connected in series and thermally connected in parallel. The required temperature reduction determines the number of couples needed in the TEC and the cooling capacity is directly proportional to the supply current in a particular package.

Several options exist for supplying power to the TEC unit. DC power sources, such as batteries, are an attractive option for providing power since it does not produce ripple effects or noise which can adversely affect TEC performance. Another source of power involves the use of pulse width modulation which is a technique for converting AC line voltage to a lower voltage DC signal. This source of power has the advantage over exclusive DC power due to unlimited AC voltage; whereas, DC power may require battery replacement.

In practice, thermoelectric coolers are used in conjunction with a heat sink and a cooling fan that provides forced air convection over the heat sink. For the application considered in this design project, the TEC would be mounted directly to a heat sink.

LIQUID HEAT SINKS

The use of a liquid heat sink (LHS) was also examined to solve the heating problem in the LCDs. Produced by the Aavid corporation, liquid heat sinks are multi-layered plastic pouches that contain a high dielectric perfluorocarbon liquid called Fluorinert. As shown in Figure 3, the inner layer of the pouch provides a barrier to ensure containment of the fluid and exclusion of air or other gases.

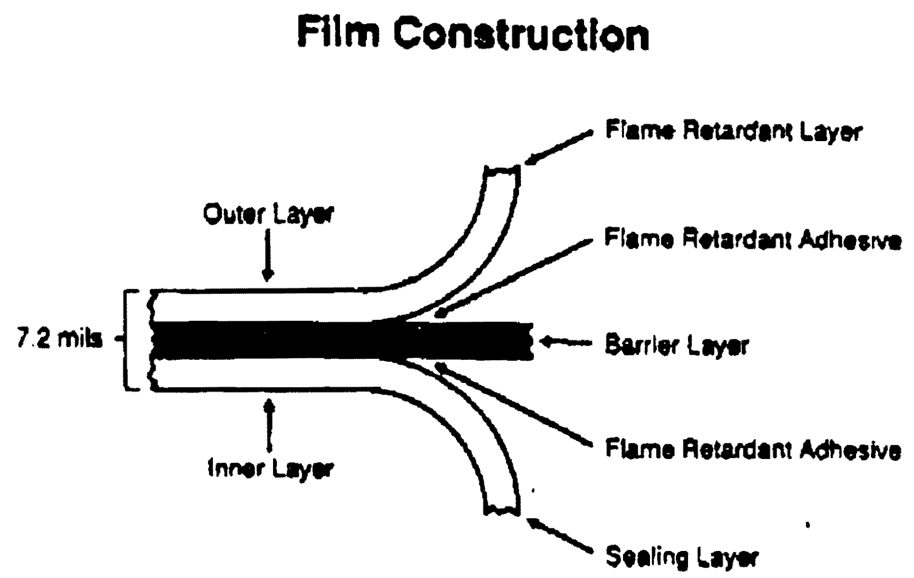


Figure 3. Diagram of the film construction of a liquid heat sink.

Heat is transported through the LHS in several stages. The first stage involves conduction from the heat source through the plastic film layer in the liquid heat sink. Next, the heat is transferred by convective currents in the fluid and is finally conducted to an external heat-dissipating surface such as a cold wall.

Liquid heat sink technology presents many advantages in its usefulness as a cooling technique. First and foremost is the relative low cost of each cooling unit. In addition to this attractive feature, Aavid is also capable of producing custom liquid heat sinks tailored to fit specific applications with tolerances of +/- 0.02 inches. For the particular problem at hand, a modified version of the existing rectangular model would have to be utilized so as to accommodate the light path through the LCD. Also, LHS have no moving parts, create no sound, and do not produce unwanted electrical signals or interference.

OPTIMIZING EXISTING FORCED FAN CONVECTION SYSTEM

A fan provides the beneficial effect of moving cooling air through an enclosure or over a target surface. Although fans are currently employed in the existing system, optimization techniques may be utilized to maximize heat transfer. Due to the unacceptable heat dissipation in the LCD component, given temperature limits govern the required performance of a forced convection fan system. Fan induced flow rate to maintain these temperatures is found with the relationship:

$$Q = fW/(Tout - Tin)$$

where Q = flow rate, cfm; f = flow factor and is the product of air density and specific heat; Tout = exhaust air temperature, deg F; Tin = ambient air temperature, deg F. At sea level, f = 3.1; at 5000 ft, f is about 3.6.

Most fans have a similar noise profile. The profile spans 63 to 8000 Hz and usually peaks at about 250 Hz. This peak gives the fan its characteristic sound or whine. Fans with the lowest sound pressure level peaks should be the main candidates. Unfortunately, not all vendors noise test their fans the same way, and reported data are often not comparable. Testing can also indicate which fans vibrate more than others. This information often does not show up in vendor data. Rotor imbalance and 60-Hz hum from a poorly made stator cause an amplified rumbling, which is more irritating than fan noise. Generally, fans with plastic housings vibrate more that those with metal housings. Also, quieter fans tend to be more expensive. For these reasons, a purchase specification should include a noise rating as well as flow rate.

Using two or more axial fans in series or parallel may be a better alternative than a larger fan. First of all, two identical fans are usually less noisy than a single larger unit. Secondly, depending upon arrangement, either static pressure or airflow may be increased while keeping the other parameter near constant. When two fans work side by side, for example, airflow should double at free delivery. However, the higher the system impedance to flow, the lower the flow increase from the second fan. Hence, a parallel arrangement is recommended when the fans operate in low impedance near free delivery. When one fan pushes air into an enclosure and another pulls air out, the fans are in series. best results from using fans in series are in systems with high impedance.

The use of a centrifugal fan is another alternative that may provide favorable results. Since the flow direction associated with a centrifugal fan is tangential to its path of rotation, higher flow rates may be possible. As a result, a higher rate of heat transfer may be obtained to cool the LCD components.

There are several benefits associated with using an axial or centrifugal fan arrangement. First, fans bought in bulk may be purchased at relatively low costs. In addition to the cost benefit, fans are simple and extremely reliable devices that have operating lives at least as long as that of the television projection set. Finally, fan use has been a proven technology for cooling electronic equipment in the computer industry and could be readily applied to the problem at hand.

CONDUCTIVE HEAT TRANSFER STRIPS

A conductive heat transfer strip was also considered as a possible solution. The strip is primarily constructed of a micro cellular urethane material called PORON. This conductive material consists of a .025 mm thick layer of copper foil wrapped around a strip of PORON.

Mounted by a layer of acrylic transfer adhesive that is applied directly, the strip also utilizes the effects of conduction by placing a highly conductive cover plate over the strip. With the cover plate in place, the strip compresses and reduces the contact resistance present between the heating source and the strip. To achieve maximum performance, the strip must at least be in direct contact with the area to be cooled. While actual values for thermal conductivity and heat transfer coefficients are not available, the conductive strip's relatively low cost is conducive to experimental testing so as to obtain meaningful results.

The major advantages of the heat strip include its ability to absorb shock and vibration, its high thermal conductivity, and its relative simplicity.

FEASIBILITY ANALYSIS

Table 1 is a ranking matrix of all the cooling alternatives studied in the background search. The various ranking values are based on seven parameters including cost, size, heat dissipation, effectiveness, noise level, reliability, and safety.

	LIQUID HEAT SINK	*THERMO ELECTRIC COOLER*	*HEAT PIPE*	*HEAT STRIP*	*ELECTRIC FAN*	*THERMO-SIPHONS*
COST	9	5	5	8	9	3
SIZE	9	10	7	7	8	5
HEAT DISSIP-ATION	6	7	9	8	7	8
EFFECT-IVENESS	6	8	9	8	8	8
NOISE	10	10	10	10	4	7
SAFETY	9	8	9	9	9	9
RELIAB-ILITY	10	8	10	10	9	7
TOTAL	59	56	59	60	54	47

Table 1. Ranking matrix for determining best cooling methods to research experimentally. Ranking: 1 - worst: 10 - best.

The calculations used to support the heat dissipation rankings are included in Appendix B of this report. These calculations represent thermal resistance models used to determine heat transfer rates and thermal resistances. In the analysis, rates of heat transfer associated with the addition of cooling systems are compared to the heat transfer rate of the current forced convection system in an attempt to show the improvements in heat flow using the cooling alternatives. The numbers calculated are approximate values and should only be considered as a means of comparison so as to derive the best method for cooling the LCD component.

PRELIMINARY TESTING SELECTION

Based on the preliminary calculations obtained from the thermal resistance models, the ranking matrix information, and the feedback from Franz Raetzer of the Philips Electronics Corporation, the three best methods for cooling the LCD components in the cabinet would be the use of an alternative forced convection system, conductive heat transfer strips, or a thermoelectric cooling device (sinked to a Liquid Heat Sink). In order to better understand the nature of the problem and gain further insights into the performance of the methods mentioned above, extensive testing and experimentation will need to be performed. These tests will, in turn, provide valuable numbers to be used in further modeling efforts.

TEMPERATURE AND FLOW MEASUREMENT DEVICES

Temperature measurements were essential in analyzing new cooling alternatives. Type T thermocouples composed of copper and constantan wires were used to test the temperature difference between the ambient air and the temperature of the liquid crystal display (LCD). Constantan wire is composed of 55% copper and 45% nickel. The thermocouples were calibrated before testing using an ice bath. These thermocouples were connected to a control panel along with an Omega digital display device, which showed the measured temperature. The experimental setup is shown in Figure 4. The thermocouples were placed on the center of each of the LCD's using masking tape. The temperature was measured within +/- 1 degree Celsius on the area of the LCD where the thermocouple was in direct contact. Temperature measurements were taken for two hours in increments of 20 minutes. Table 2 provides the actual data obtained in the laboratory from the tests run on the original forced convection system.

LCD	t = 0 min	t = 20 min	t = 40 min	t = 60 min	t = 80 min	t = 100 min	t = 120 min	Delta T
Red	26.5 C	30.8 C	32.1 C	35.9 C	36.8 C	37.4 C	37.6 C	11.1 C
Green	26.0 C	33.2 C	34.4 C	35.6 C	37.0 C	38.2 C	38.5 C	12.5 C
Blue	25.8 C	41.5 C	43.4 C	44.4 C	45.7 C	46.4 C	47.0 C	21.2 C

Table 2. Actual experimental results for baseline tests run on the LCD components with no additional use of cooling techniques.

A plot of these results is shown in Figure 5. From the graph, the temperature of each LCD began to approach steady state temperature as the time increased.

The air flow through the use of fans was tested using a mini-anemometer. The anemometer uses a **Kurz** low-power sensor, which consists of a temperature and velocity sensor. The velocity sensor is heated and operated as a constant-temperature thermal anemometer that responds to a standard velocity by sensing the cooling effect as air passes over the heated velocity sensor. The temperature sensor compensates for a wide range of ambient temperature variations. The velocity readings are referenced to a standard temperature of 25 degrees Celsius and a barometric pressure of 760 millimeters of Mercury. The anemometer exhibits extraordinary sensitivity and resistance to shock and vibration due to its coil configuration. The basic sensing element of the anemometer is a probe containing both a velocity and temperature sensor. The probe shaft is made of a non-conductive graphite that has a velocity range up to 12,000 feet per minute. To measure the air flow from the fans, the probe was placed in the direction of air flow and the meter indicated the resulting velocity.

Figure 4. Temperature Measurement Set-up

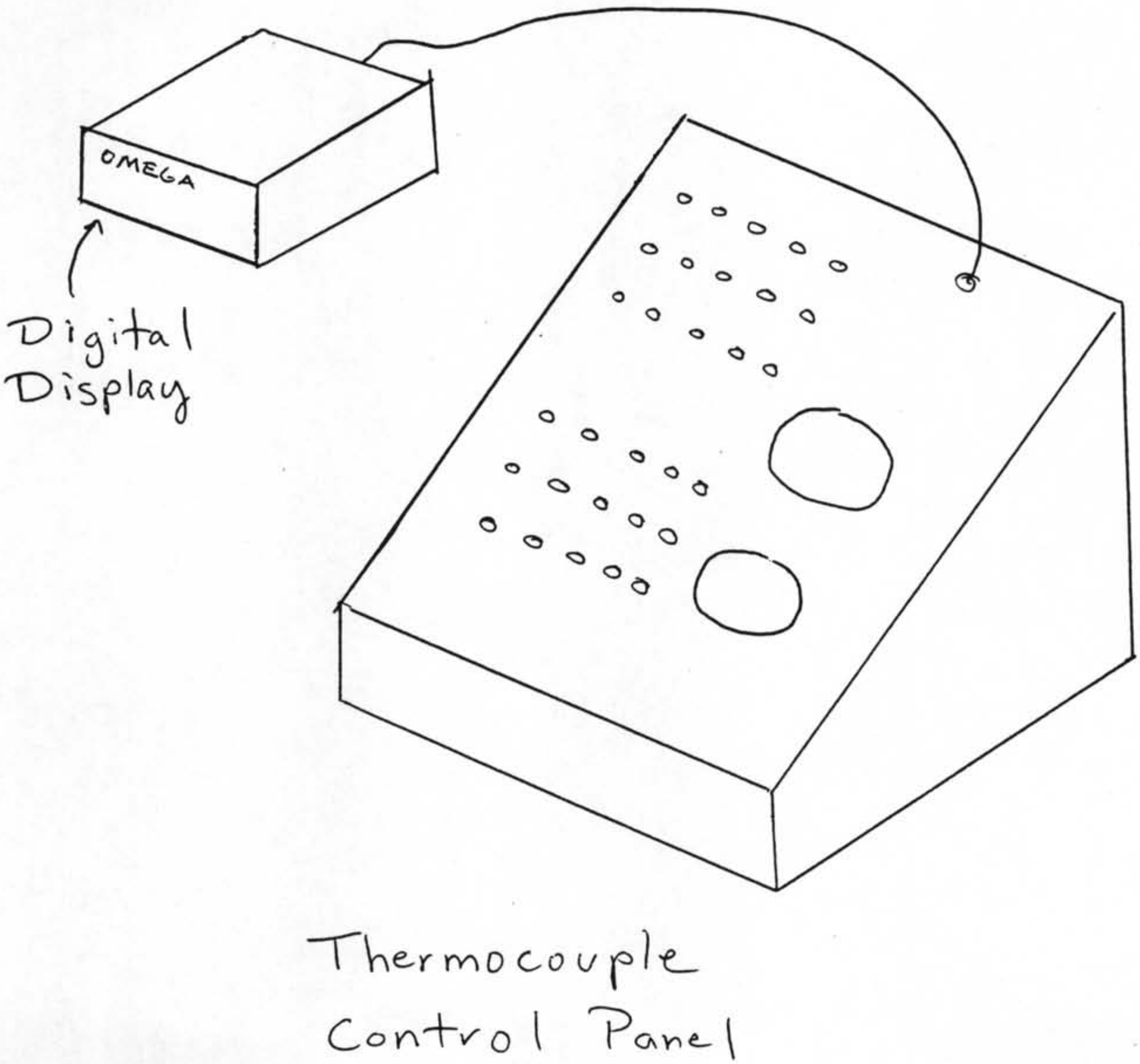
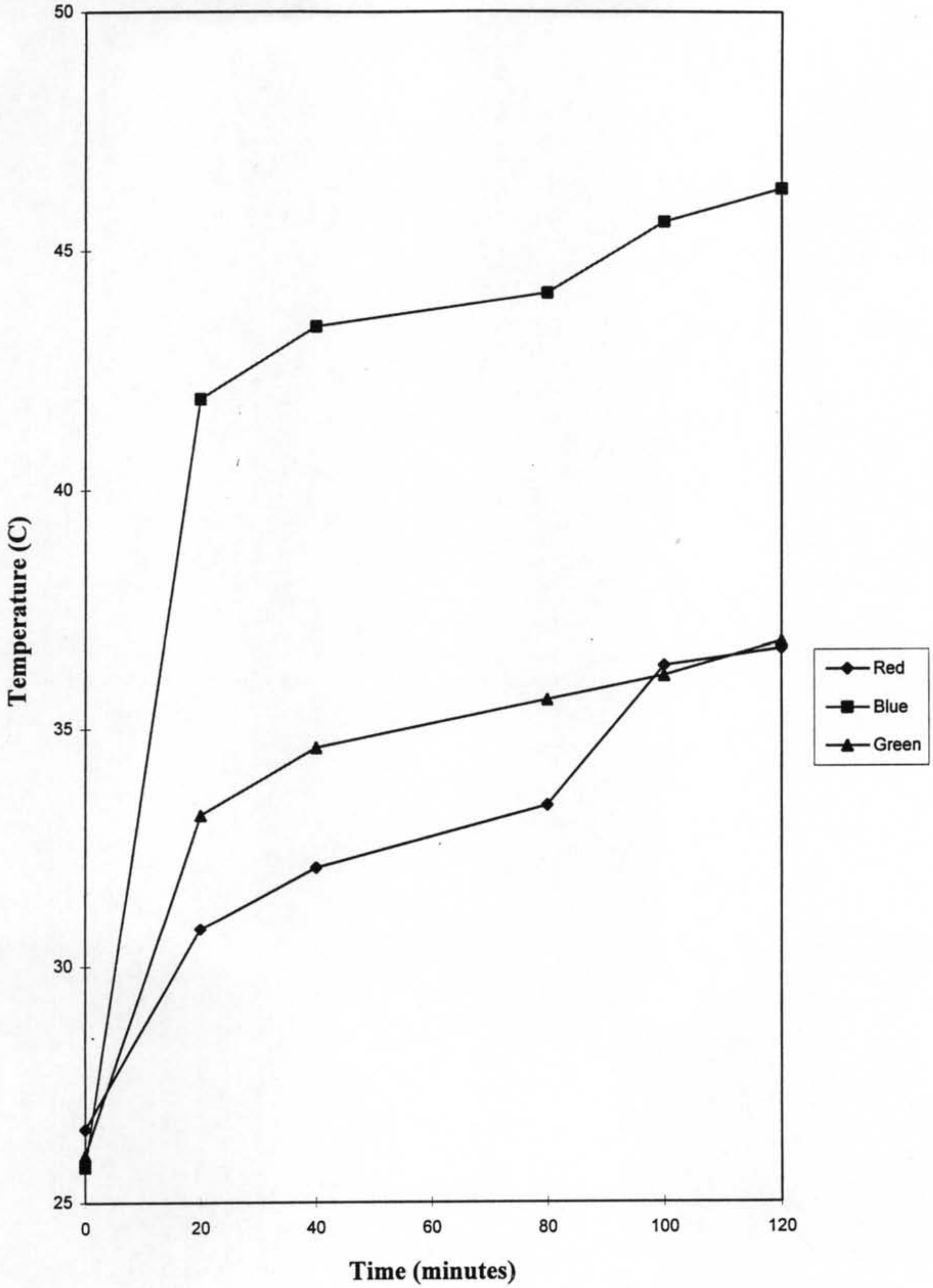


Figure 5. Time Vs. Temperature of Original Forced Convection System



TESTING OF SELECTED COOLING TECHNIQUES

The methods of cooling the LCD components include the use of a thermoelectric cooler, conductive heat transfer strips, liquid heat sinks, and an improved forced convection system. The following sections describe the initial experimental setup, the procedure followed in testing, the results produced by the tests, and an explanation of the experimental findings. In addition to the results obtained in the laboratory, a sketch of the testing geometry is included in order to clarify the placement and orientation of each proposed cooling technique. Also, performance plots were obtained to show the transient response of the LCD temperature over time until a nearly steady-state condition was achieved.

The central goal of this testing was to analyze the temperature measurements in terms of temperature differences. With this type of temperature measurement scheme, the initial temperatures of the LCD components were measured and recorded. Then, after taking measurements at 20 minute increments, the total temperature difference between $t = 0$ minutes and $t = 120$ minutes was calculated. Consequently, the temperatures discussed in this report do not necessarily represent the actual temperatures of the LCD components. However, by comparing the temperature differences associated with each LCD in the situation of the original forced convection system to the newly proposed cooling methods, a quantitative assessment of the benefits or detriments of the new system could be ascertained.

Conduction Testing

The first alternative studied by experimental testing involved the use of copper tape. As shown in Figure 6, the lens attached to the LCD frame was initially removed. Once this had been done, thin strips of copper tape were mounted directly to the aluminum frame of the LCD itself in an attempt to increase the conduction path from the center of the LCD to the outer edges. Also, two T-type thermocouple wires were taped directly to the center of each LCD using ordinary masking tape as explained in the temperature measurement portion of the report. Then, the lens was replaced on the LCD housing covering the tape and the newly applied thermocouple wires. To begin the test procedure, initial temperature measurements of the surface of the LCD were taken and averaged for each LCD. Then, allowing the television to operate for up to two hours at a time, data was collected at 20 minute intervals in order to ensure the validity of the readings. Table 3 shows the results obtained in the laboratory.

LCD	t = 0 min	t = 20 min	t = 40 min	t = 60 min	t = 80 min	t = 100 min	t = 120 min	Delta T
Red	24.9 C	30.9 C	32.6 C	35.7 C	37.1 C	38.3 C	38.9 C	14.0 C
Green	23.8 C	34.6 C	36.3 C	37.6 C	39.3 C	40.3 C	41.0 C	17.2 C
Blue	24.3 C	43.6 C	45.7 C	47.9 C	47.9 C	48.7 C	49.6 C	25.3 C

Table 3. Experimental results obtained with the use of copper tape mounted on the LCD components.

Once the television had reached an approximate steady-state temperature (after approximately 2.0 hours), the final reading was taken and the "actual" temperature was plotted versus time as shown in Figure 7. From this plot, a direct comparison could be made between the temperature of the LCD components with the copper tape and the components without the tape as discussed in the section concerning temperature measurements. As can be seen from

Table 4, the use of copper tape adversely affected the temperature of the LCD components by as much as a 4.7 degree Celsius increase.

LCD Component	Delta T without copper tape	Delta T with copper tape	Percent Increase in temperature
Red	11.1 C	14.0 C	20.7 %
Green	12.5 C	17.2 C	27.3 %
Blue	21.2 C	25.3 C	16.2 %

Table 4. Increase in temperature involved with using the copper tape in the experiment. This increase, consequently, shows that the tape does not benefit the system.

An explanation for this discrepancy could be that the model did not sufficiently take into account the contact resistances between the aluminum frame of the LCD and the copper tape. Consequently, since the tape affected the performance of the current conduction path in a negative manner, it would not be suggested as a viable means for cooling the LCDs.

The second alternative scrutinized by experimental testing was the use of *conductive heat transfer strips*. These strips distributed by the Rogers Corp. essentially are rectangular pieces of a material named Poron (a foam-like material) surrounded by a shell of copper. One side of these strips has an adhesive that can be used to mount the strip on a particular surface. In the experimental arrangement shown in Figure 8, the conductive strips were placed on the LCD frame similarly to that of the copper tape. Then, following the same experimental procedure as that for the copper tape measurements, temperature data was collected over a period of approximately two hours as shown in Table 5.

LCD	t = 0 min	t = 20 min	t = 40 min	t = 60 min	t = 80 min	t =100 min	t =120 min	Delta T
Red	20.6 C	36.3 C	39.7 C	39.4 C	39.4 C	40.3 C	40.7 C	20.1 C
Green	21.1 C	42.2 C	45.6 C	45.1C	45.2 C	46.2 C	46.8 C	25.7 C
Blue	21.1 C	47.5 C	51.2 C	51.0 C	51.0 C	52.1 C	52.5 C	31.4 C

Table 5. Experimental results obtained in the lab with the conductive heat transfer strips mounted on the LCDs.

From Figure 9, another plot of the apparent temperature versus time could be compared to the baseline measurements obtained in the temperature measurement portion of the report. From this comparison, an even greater temperature difference was obtained from the use of conductive heat transfer strips than for the copper tape measurements. As shown in Table 6, the temperature difference of the system with the heat transfer strips increased by as much as 13.2 degrees Celsius over the values obtained in the baseline testing.

LCD Component	Delta T without Conductive Strip	Delta T with Conductive Strip	Percent Increase in temperature
Red	11.1 C	20.1 C	44.8 %
Green	12.5 C	25.7 C	51.4 %
Blue	21.2 C	31.4 C	32.8 %

Table 6. Increase in temperature involved with using the conductive heat transfer strips in the experiment.

This result is due to the fact that the strips themselves block nearly all airflow from the existing forced convection system. Consequently, the temperature increases at a substantially higher rate when the heat transfer strips have been applied. Unfortunately, the manufacturer of the Poron strips has informed us that the size of strip used in these experimental tests are the smallest version produced. As a result, the use of the conductive heat transfer strips has been eliminated from the list of viable cooling methods for this project.

Figure 6. LCD Housing With Lens Removed

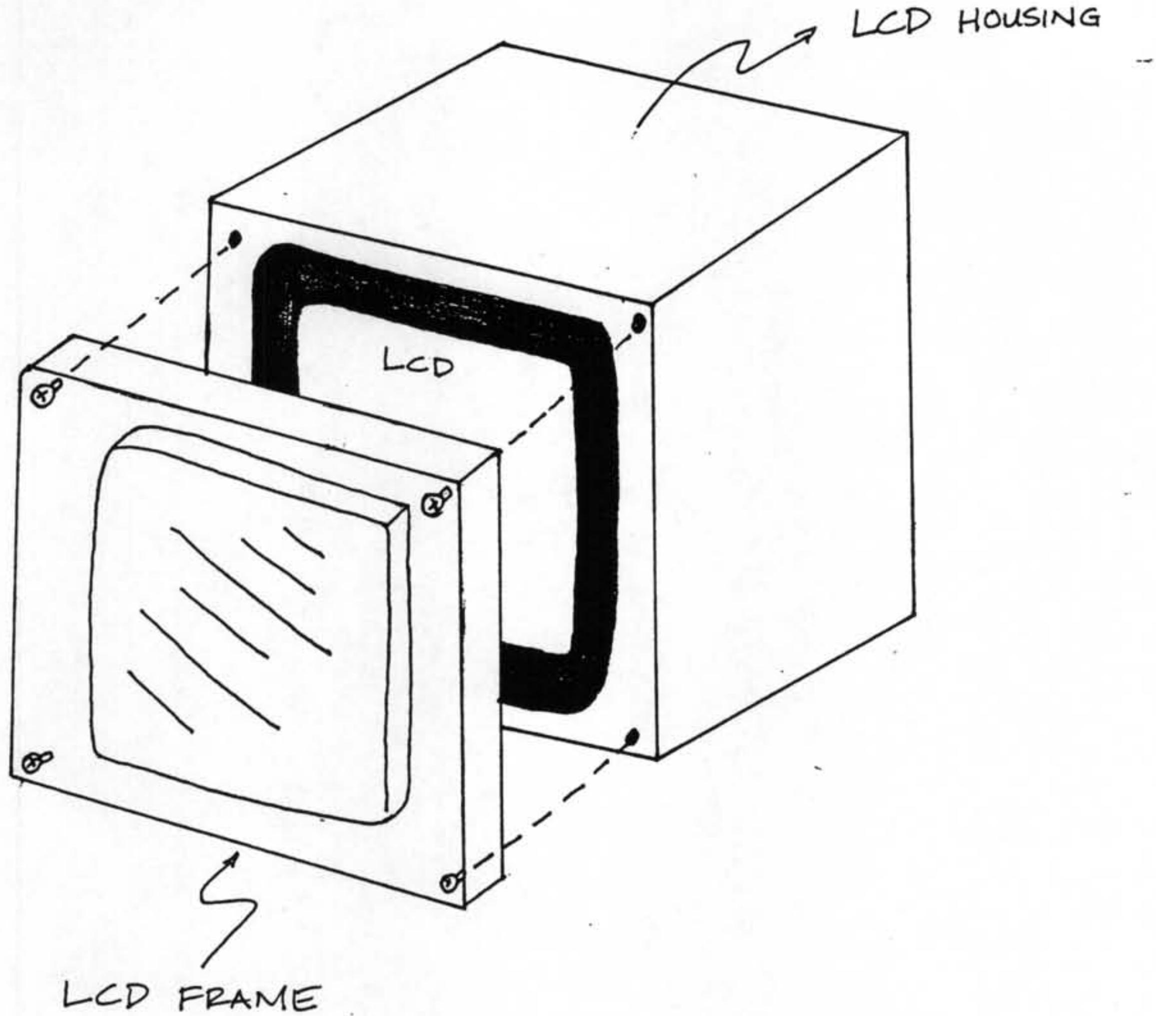


Figure 7. Temperature Vs. Time Associated with Copper Tape Addition

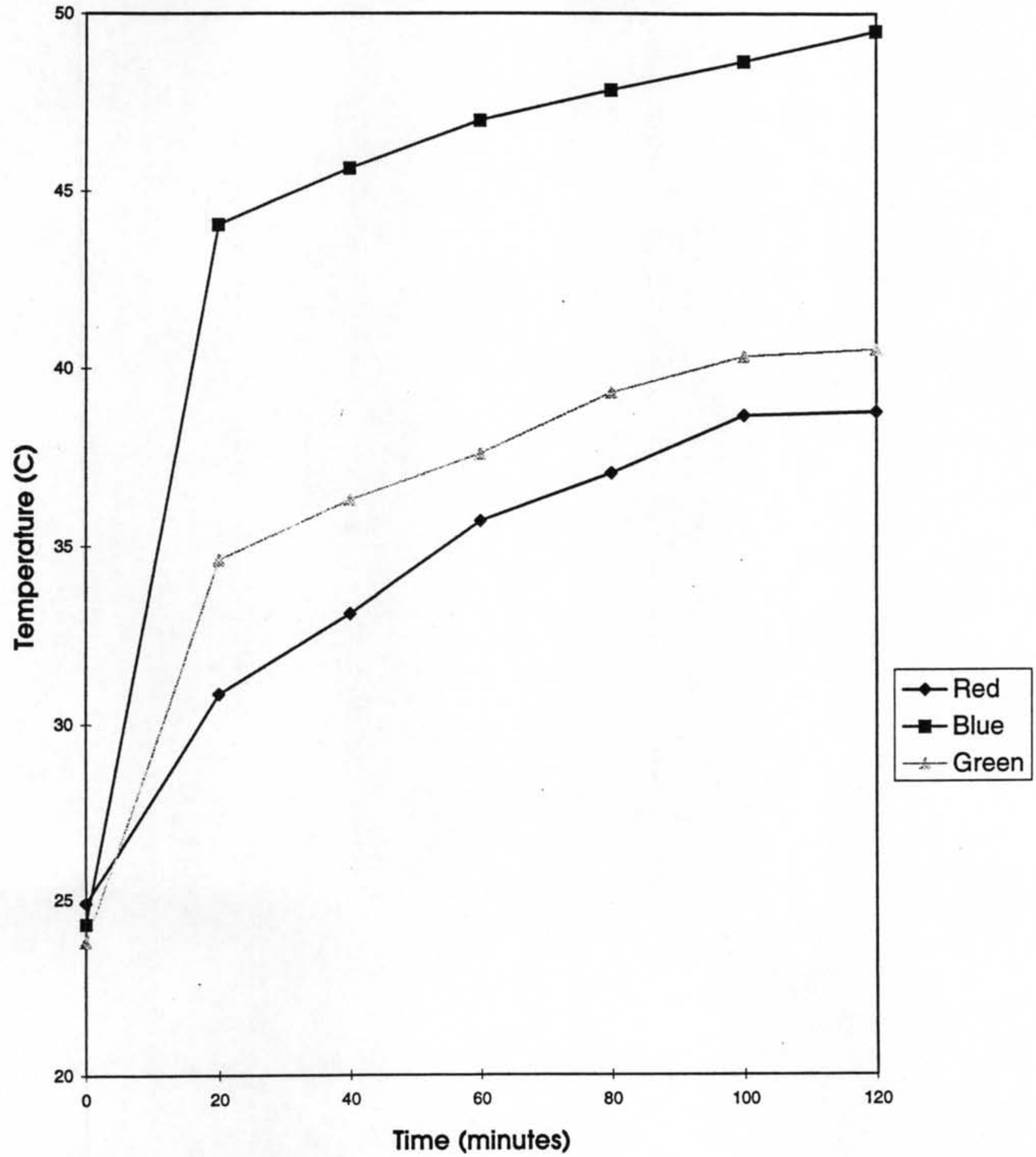


Figure 8. Placement of Conductive Heat Transfer Strips

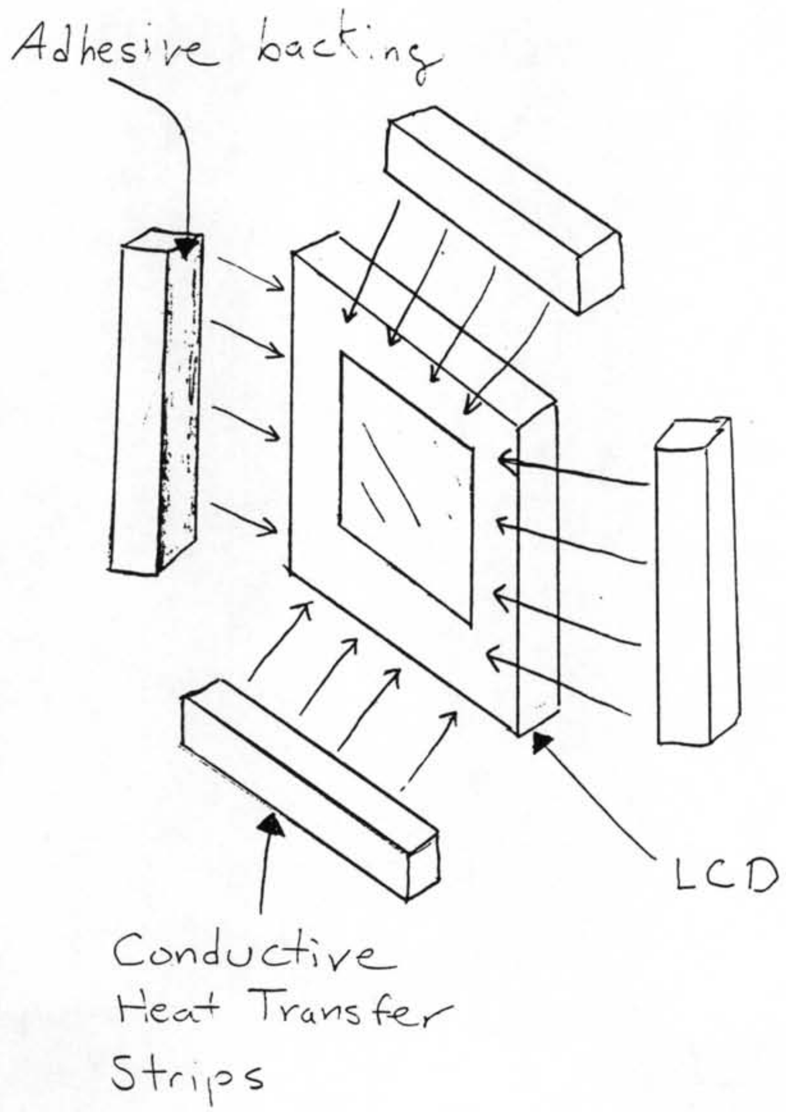
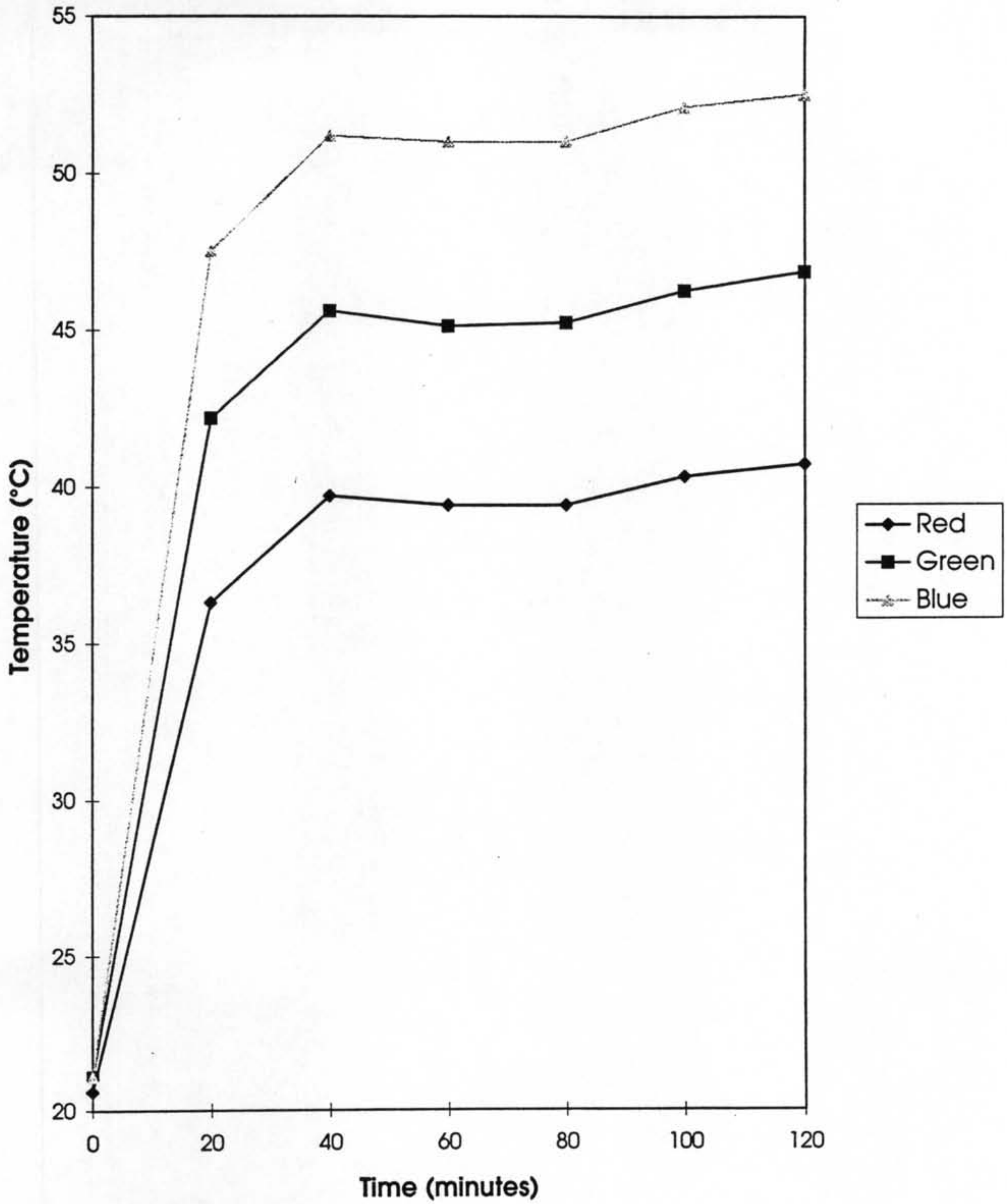


Figure 9. Temperature Vs. Time Plot Associated with Conductive Heat Transfer Strips



Thermo-electric Cooler and Liquid Heat Sink Testing

Three alternative techniques were applied and required temperature measurements to be recorded using the same procedure described in the temperature measurement portion of the report. There were a total of four tests which were done. The first test was performed to serve as a baseline temperature measurement and was conducted solely for comparison purposes. The procedure and explanation of each test configuration is explained in the following paragraphs.

TEST 1: Axial Fan Configuration

This test was described in detail in the temperature measurement portion of this report. It involved the determination of the temperature difference between pre-start up and steady- state times. Table 7 displays the experimental results obtained in the laboratory.

LCD Component	t = 0 min.	t = 36 min.	t = 60 min.	t = 70 min.
RED	23.6 °C	32.6 °C	33.2 °C	33.4 °C
GREEN	23.4 °C	24.7 °C	24.8 °C	25.0 °C
BLUE	23.8 °C	41.3 °C	41.6 °C	41.5 °C

Table 7. Baseline temperature measurements using current axial fan configuration

TEST 2: TEC with Copper heat strip

For test 2 as shown in Figure 10, a thermo-electric cooler was placed between the LCD block and the aluminum chassis. By means of conduction, the TEC was expected to cool the LCD housing which would therefore cool the surface of each LCD. Measurements were repeated in the same manner as the first experiment except that temperature was only recorded for the first 40 minutes. The test was stopped at this time because the temperature difference associated with the LCDs exceeded the temperature difference found in the baseline measurements. These results can be found in Table 8.

LCD Component	t = 0 min.	t = 20 min.	t = 40 min.
RED	21.8	32.1	32.6
GREEN	21.3	23.4	23.1
BLUE	22	38.3	38.6

Table 8. Temperature measurements for TEC & copper heat strips.

TEST 3: TEC with Aluminum Heat Sink

The TEC was placed directly below the axial fan for this test. The TEC hot side was sinked to a finned Aluminum heat sink and was fixed to the opening of the air inlet section. A more precise picture of the configuration is shown in Figure 11. The TEC cold side was directly under the axial fan and was used to cool the inlet ambient air. Table 9 provides the experimental results for this test run on the LCD.

LCD	t = 0	t = 10	t = 20	t = 30	t = 40	t = 50	t = 60
Red	19	28.75	31	32.4	33	34	34.1
Green	19	23.2	25.4	26.45	27.6	28.25	28.4
Blue	19	33.85	36.6	37.7	38.7	39.8	39.9

Table 9. Temperature measurements for TEC & finned aluminum heat sink.

TEST 4: TEC with Liquid Heat Sink

This test configuration is shown in Figure 12. For this test, conduction cooling was again tried, but this time a liquid heat sink was used instead of a metal heat sink. The TEC cold side was adhesively attached to the metal block directly above the housing for the green and blue LCD. Temperature measurements were taken at ten minute intervals for a total of sixty minutes.

LCD	t = 0	t = 10	t = 20	t = 30	t = 40	t = 50	t = 60
Red	22.4	27.6	28.9	29.8	30.1	30.4	30.5
Green	22.6	25.75	27.35	27.55	27.7	28.05	28.4
Blue	22.6	36.2	38.1	38.9	38.9	39.3	39.5

Table 10. Temperature measurements for the TEC/LHS combination.

The results of the four tests revealed very similar results for the temperature readings. Among the four tests, the best results were obtained from using the TEC and copper heat strip combination. By positioning the TEC under the blue and green LCD channel, all three of the LCD temperatures were lowered. The temperatures were (0.8, 1.9, and 2.9)°C lower for the red, green, and blue LCDs, respectively. For test 3, the temperatures of the red and green LCDs were higher than test 1. However, the blue LCD temperature was 1.9 °C lower. Test 4 showed the temperatures of the red and blue LCDs to be lower by 2.9 °C and 2 °C, respectively. The lowest temperature measured for the blue LCD was 38.6 °C from test 2.

A graph of the temperature results are shown for tests 2-4 as Figures 13-15, respective of each test. On all three of the graphs, it can be seen that temperature rise is the greatest during the first 20 minutes of testing. After 20 minutes, the temperature increase begins to level off for each test configuration.

Figure 10. TEC with Copper Heat Strips

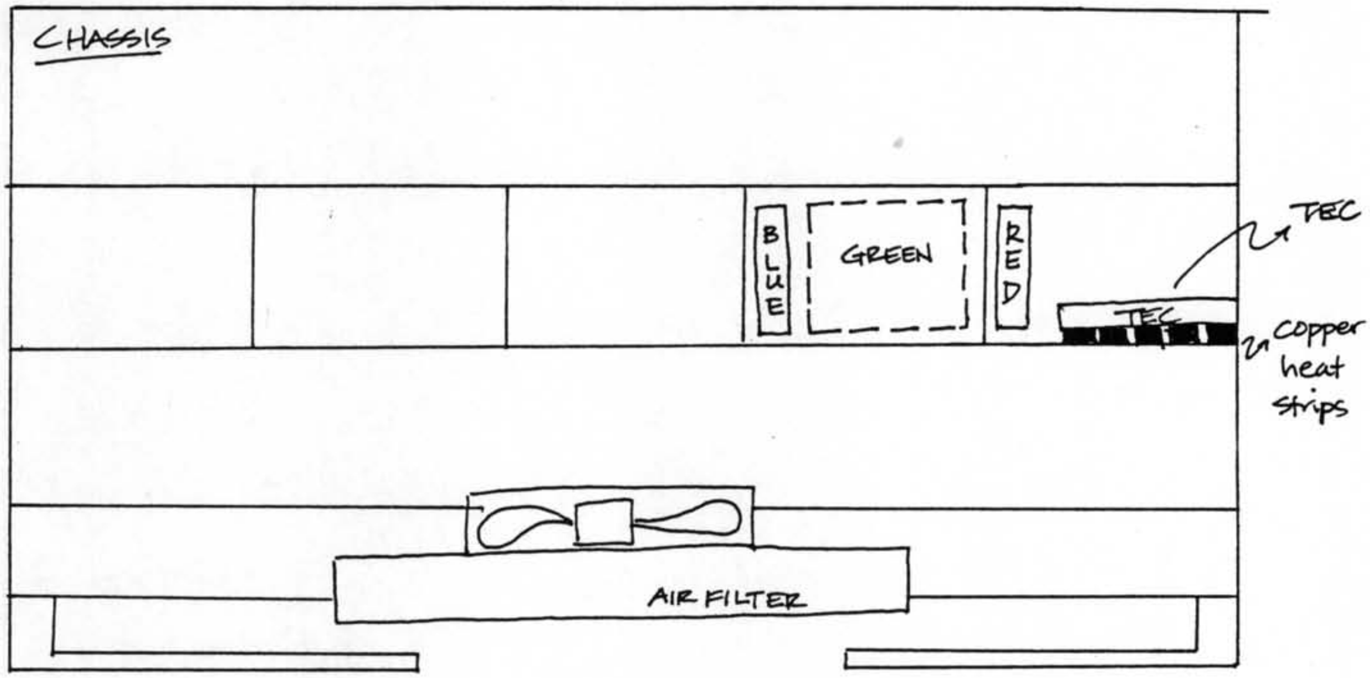


Figure 11. TEC / Aluminum Heat Sink Configuration

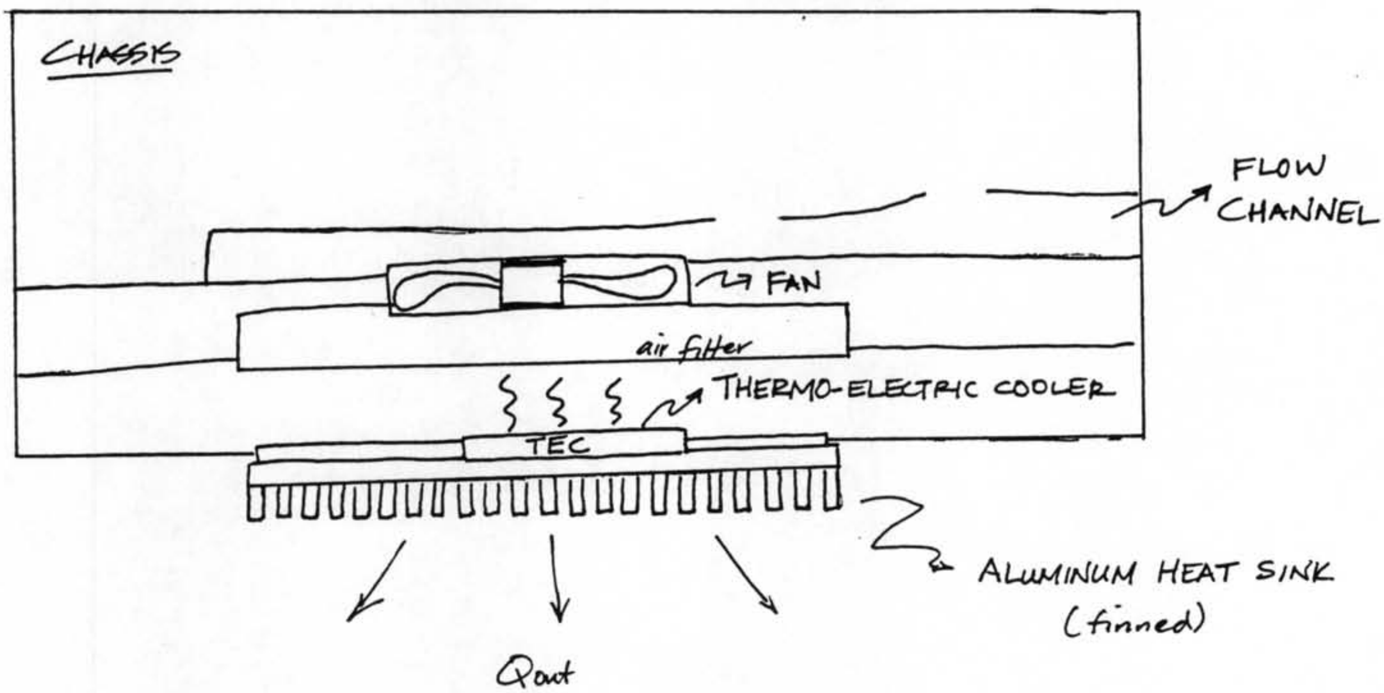


Figure 12. TEC / Liquid Heat Sink Configuration

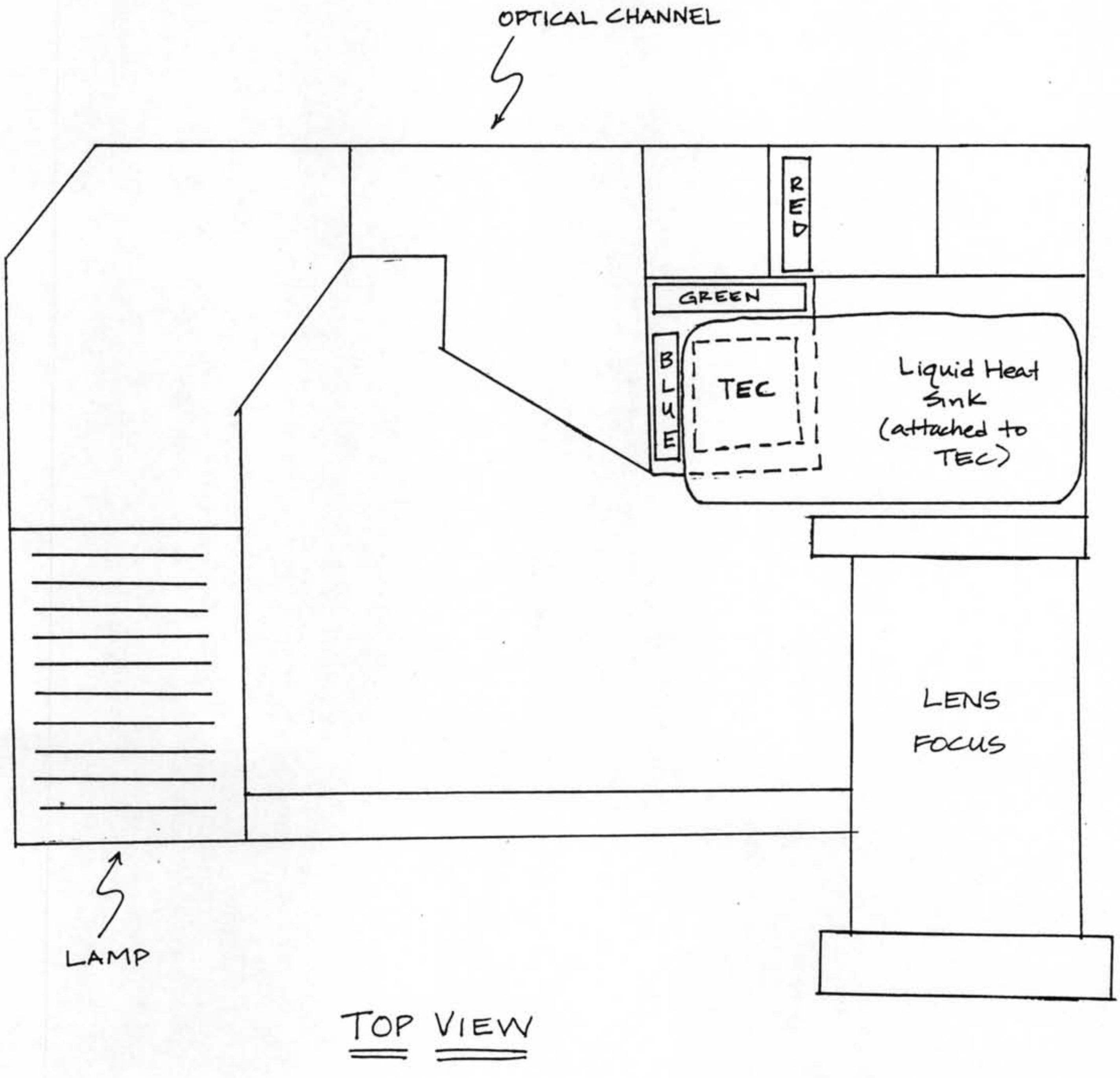


Figure 13. TEC Performance Using Copper Tape

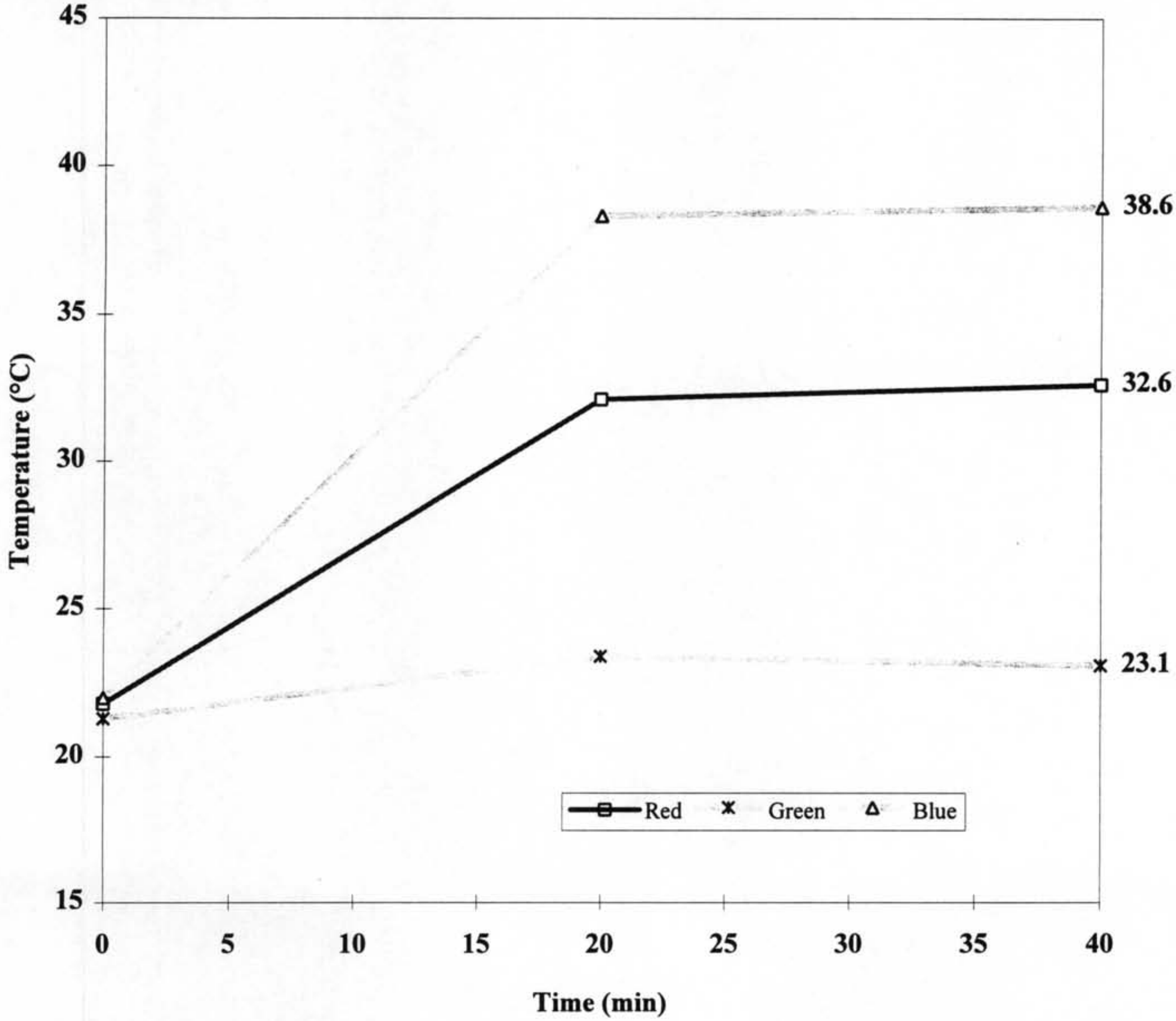


Figure 14. TEC / Aluminum Heat Sink Performance Plot

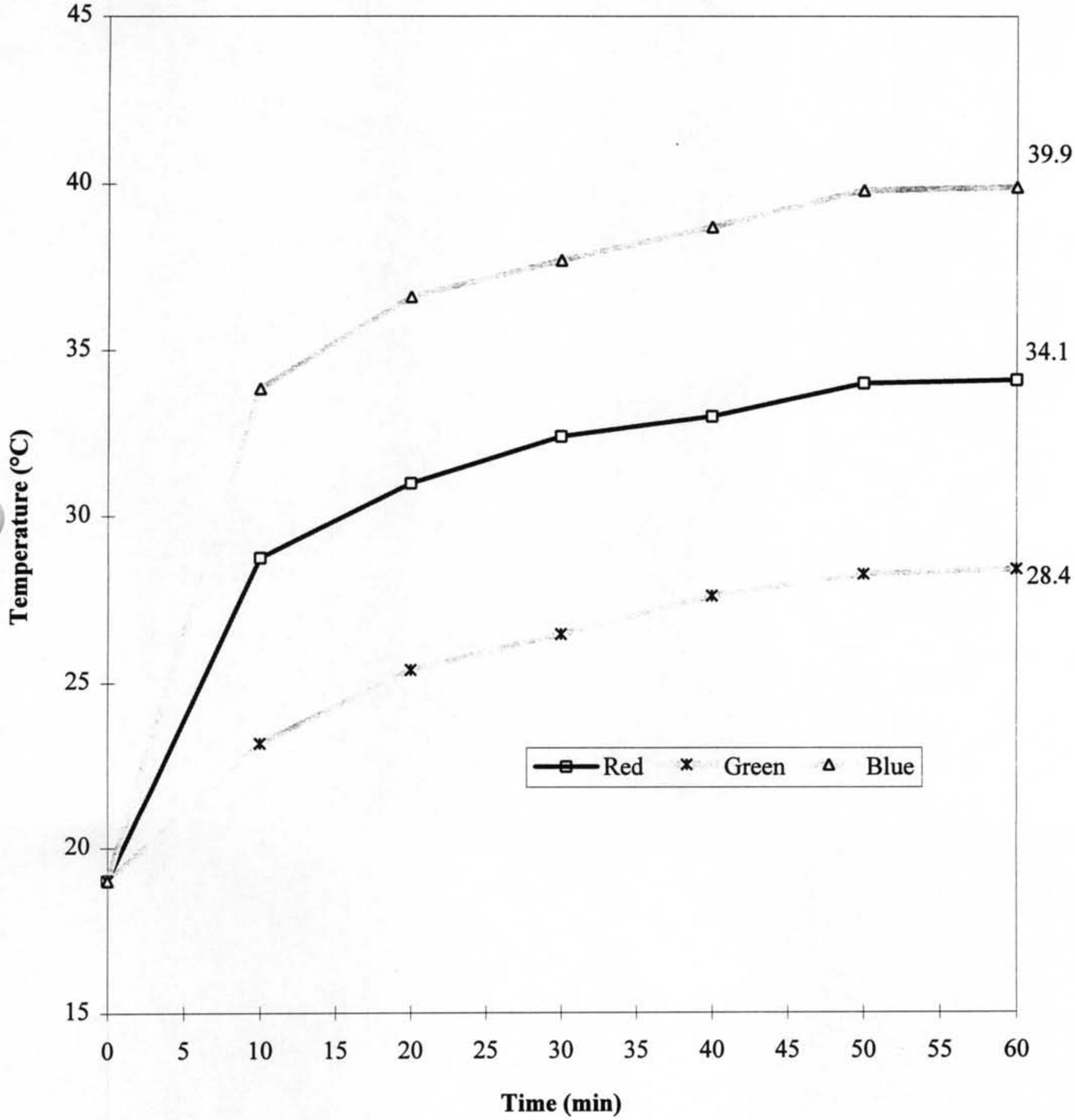
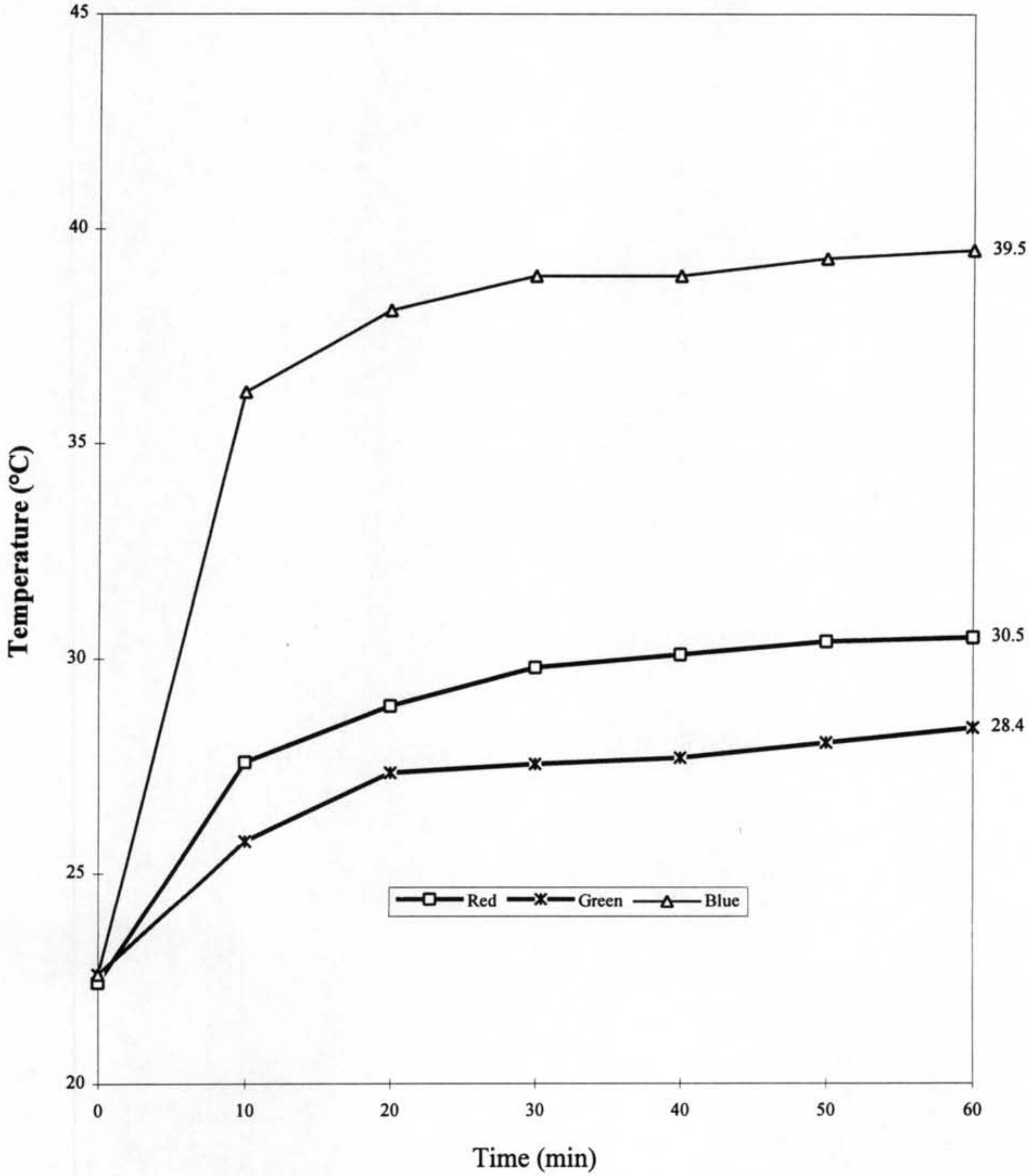


Figure 15. TEC / Liquid Heat Sink Performance Plot



CENTRIFUGAL AND AXIAL FAN TESTING

Upon inspection of the existing forced convection system and channel placement, it was decided that some preliminary testing be done to investigate alternatives to fan positioning relative to the channels. The current system consists of an 80mm axial fan blowing over the three designated channels, i.e. red, green and blue. The existing fan model and channel arrangement was used to conduct the following tests with an actual scale model built to simulate a portion of the forced convection system. For each of the experiments described below, three trials of the same test were run to validate the findings (Appendix C). Then, these three trial results were averaged.

As shown in Figure 16, the first test conducted included the existing axial fan in the built apparatus to draw some baseline values for all other testing to be done. The entire case was enclosed to eliminate any air leakage. The axial fan was placed on a mounted base in the center of the box. Results were compiled and recorded. The same test was conducted again using another measuring instrument to check the sensitivity. Averages were recorded and a baseline was established.

The second test conducted introduced the use of a centrifugal fan taken from a Con-Air brand mini hair dryer (Figure 17). A hole 2 in diameter was cut into the model to allow a draw of fresh air into the channel system. One-half inch pieces of the model material were cut and applied to the top portion of the channels to offset the extra height. Results were obtained to show an overall increase in each channel tested by an average of 350 ft/min.

The next test conducted involved the combination of two centrifugal fans maximizing the flow over the two problem channel areas, the blue and green channels. Figure 18 is provided to show the arrangement. Two similar fans from Recotron brand hair dryers (actual fans provided from Johnson controls, Athens, TN) were used in this test. The fans were mounted to wax-coated cups, with their edges flared out to maximize the total width of the channel. The 2 in. hole for drawing in fresh air was closed off during this testing. The results gained over this test were recorded with a minimum/maximum air flow measurement. An average value is provided for comparison of baseline values. The same dual centrifugal fan test was conducted but this time and alternate cup was placed over the blue channel. The channels for red and blue were switched, but the fan arrangement remained the same. The highest values ever tested were found here with air flow measurements well over 3000 ft/min taken over the blue channel.

The next test used a combination of the current 80mm axial fan and a centrifugal fan located under the blue channel as shown in Figure 19. A plastic sleeve was taped around the base of the axial fan to force the convection from the fan upward. The same measures for the omission of the 2 in. hole in the side of the model were taken for this testing also. The results showed a tremendous drop in air flow over the red and green channels in comparison to centrifugal fan usage.

The next dual fan apparatus, shown in Figure 20, used two current production axial fans. The same sleeve arrangement was present with an increased voltage usage of 14V instead of the 10-12 V range seen with the previous testing. The results rendered the poorest results so far. The greatest range of values fell over the blue channel ranging from 50 - 900 ft/min over the total surface area of the channel.

The final test administered used a 120mm axial fan similar to the current one used underneath the light assembly in the projection unit. As shown in Figure 21, the same sleeve arrangement was used but larger wax cups were cut around the base and taped together to direct air flow. The results found similar if not lower readings than the baseline values seen with the preliminary studies.

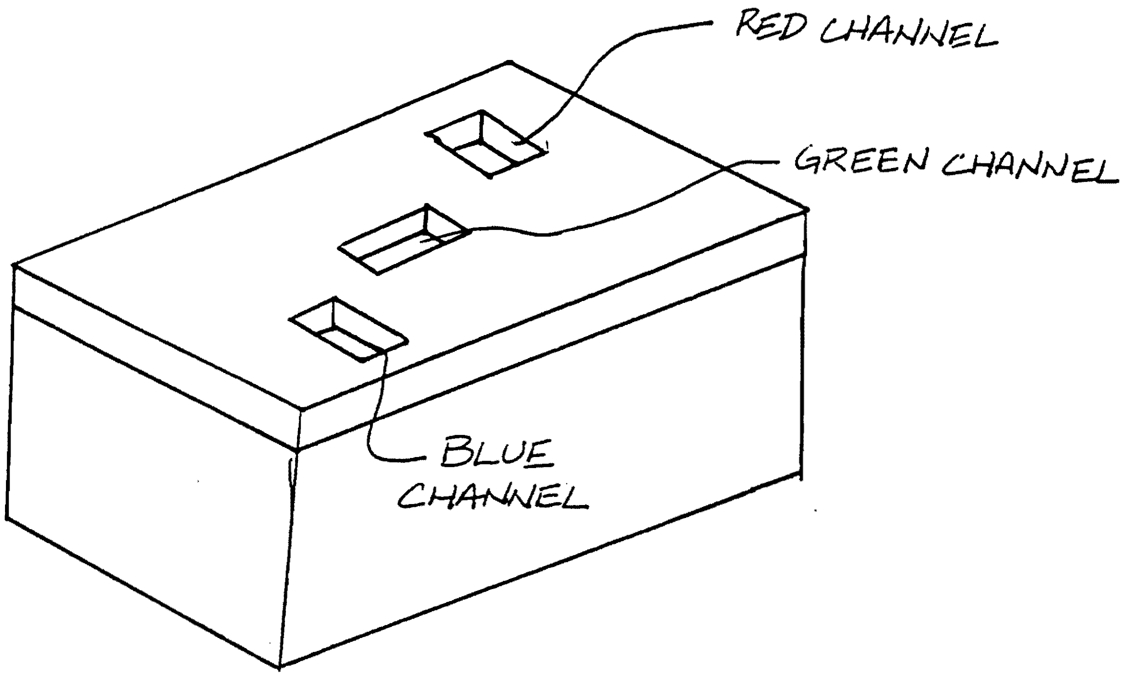
Figure 22 provides a graphical comparison of each experimental test run on the LCD and the average flow rates for each channel. The results from each of the tests described above are shown in Table 11.

LCD	Base-line	Centrifugal Fan	Dual Centrifugal Fans	Dual Centrifuagal #2	Centrifugal and 80 mm Fan	Dual 80 mm Fans	120 mm Fan
Red	550	1000	1258.33	1208.33	425	425	250
Green	546.7	866	1491.67	1075	391.67	650	300
Blue	816.7	1216.67	1233.33	2783.33	2983.33	550	925

Table 11. Average values for flow rates for each of the experimental procedures described in this section. All flow rates are expressed in ft/min.

Although the dual-centrifugal fan arrangement provided the best results when compared to all tests run on the LCD components, further testing and investigation of the results is required so as to obtain a quantitative measure of the overall effectiveness of this cooling alternative.

Figure 16. Apparatus For Baseline Fan Testing (Axial)



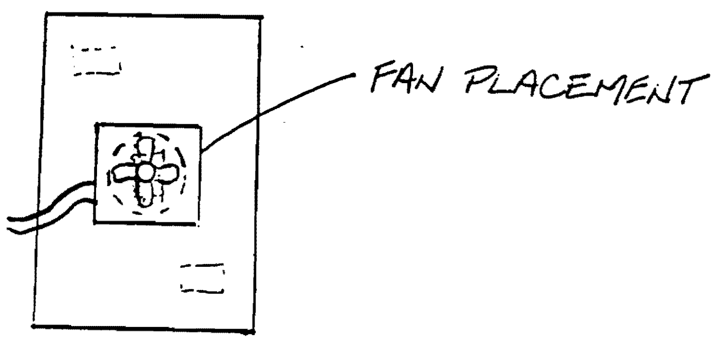
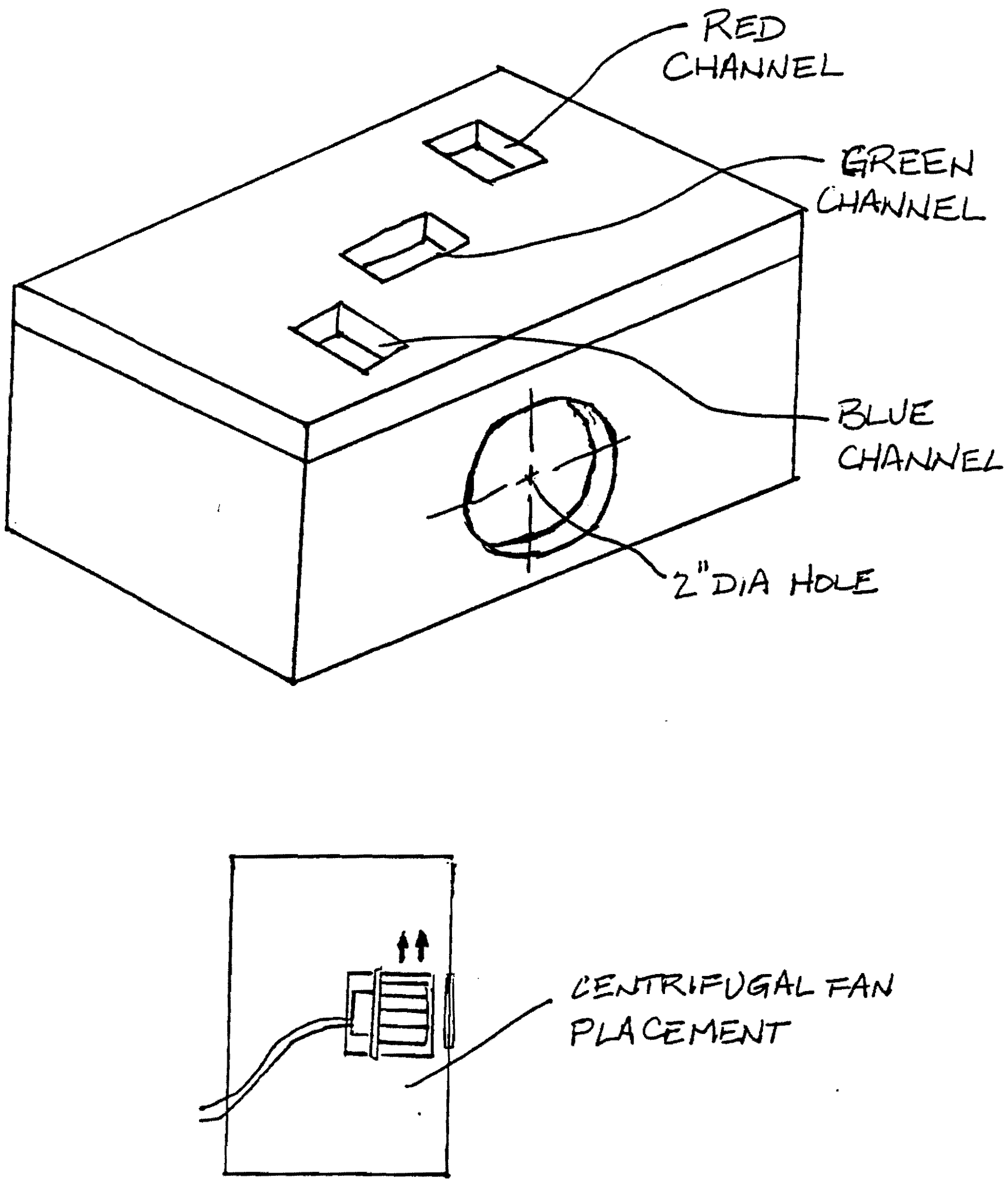


Figure 17. Centrifugal Fan Testing Orientation



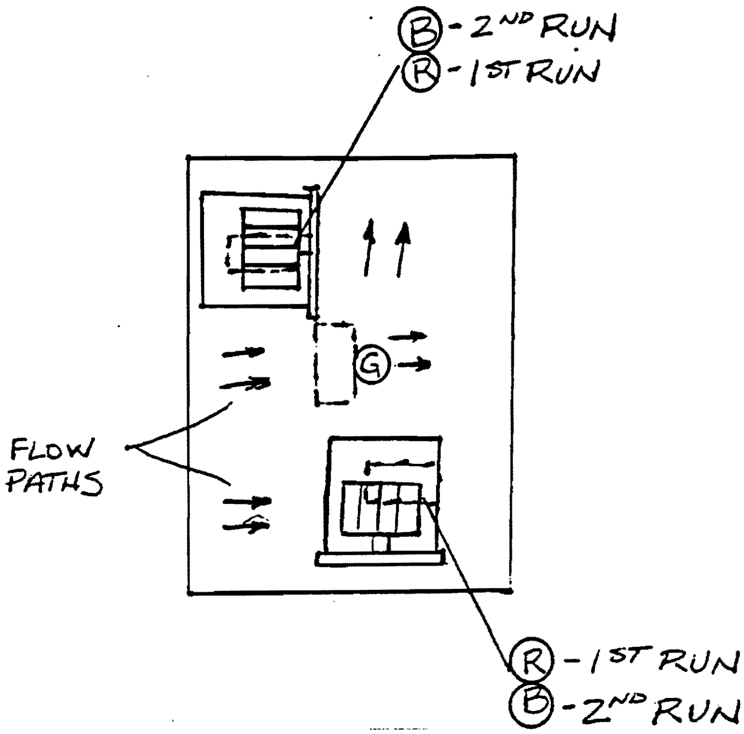


Figure 18. Dual Centrifugal Fan Placement

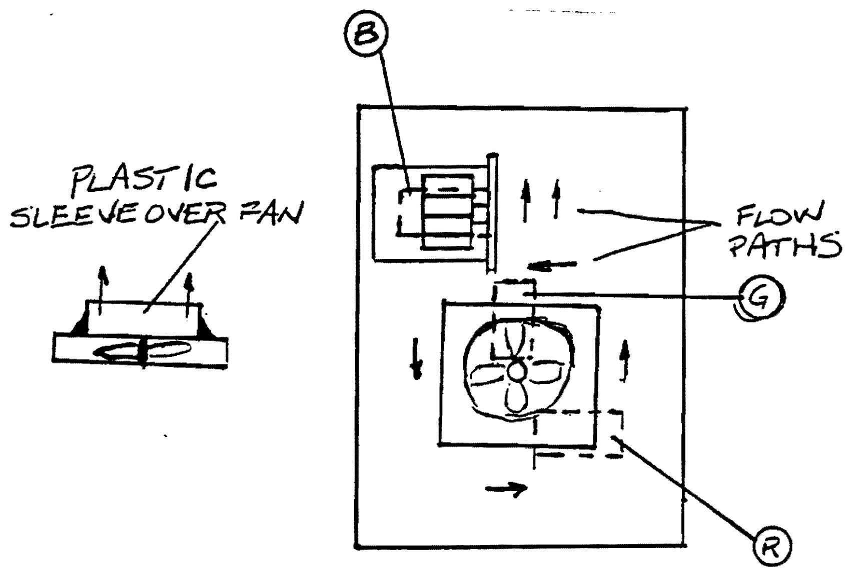


Figure 19. Combination of 80mm Axial Fan and Centrifugal Fan Placement

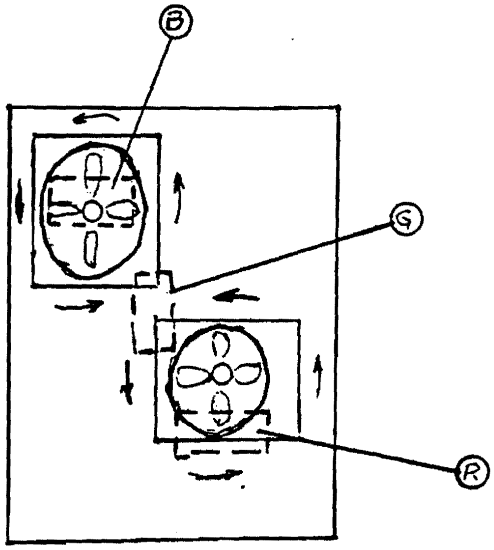


Figure 20. Dual 80mm Axial Fan Placement

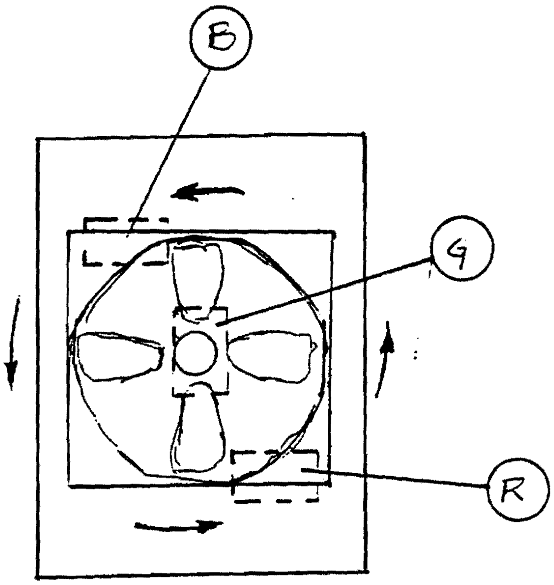
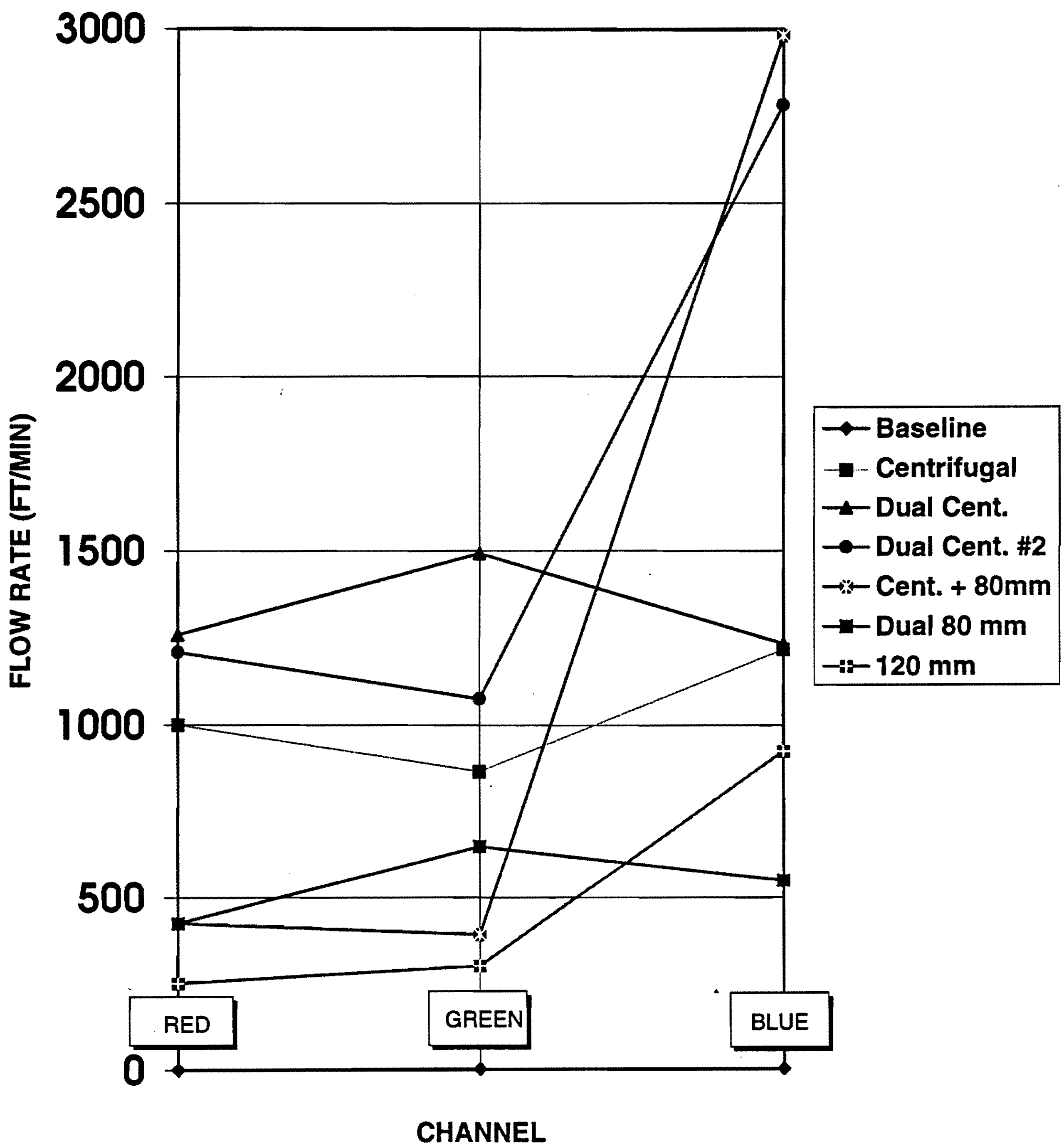


Figure 21. 120mm Axial Fan Placement

Figure 22. Average Fan Flow Rates



SELECTION OF FINAL DESIGN

Based on the experimental data obtained from the testing procedures described in the previous sections of this report, the dual-centrifugal fan arrangement was selected to be an integral part of the new cooling system. In addition to this new forced convection system, the use of a thermoelectric cooler was also used in conjunction to the dual centrifugal fan setup as a possible means for cooling the LCD. With these two alternatives working together, we anticipated a functional cooling system that would meet the requirements of the design criteria.

FINAL DESIGN CONFIGURATION

Experimental Setup

The final design configuration as selected from initial testing and preliminary calculations results consists of a dual centrifugal fan arrangement and the use of a thermoelectric cooling device. Simplification of the existing flow path was attained by moving the fans underneath the LCD components, thereby, allowing air to move across the heat source directly. Figure 23 is a schematic diagram of the experimental setup. An equipment list for the tests described in this report are listed in Appendix D. Since the existing flow channels could not be used in testing, a mock-up of the new flow configuration was constructed. This model was constructed with sheet metal by R & M Climate Control. A two inch hole on one side of the "box" was created for air intake was machined into the model. As shown in Figure 24, the model has the same hole arrangement as that of the existing flow channels with one variation. This variation as mentioned above is the positioning of the centrifugal fans directly below the LCD components. The tests conducted involved the combination of two backward curved centrifugal fans maximizing the flow over the two problem channel areas, the blue and green channels. These fans were obtained from Comair-Rotron brand hair dryers. In order to direct the air flow from each centrifugal fan, two housings were fabricated. These housings as shown in Figure 25 were cylindrical in shape with an opening cut at the top to allow the air to move in a tangential direction from the fan blades. As mentioned in the background information portion of this report, thermoelectric coolers must be attached to a heat sink on its hot side. For this reason, sheet metal was selected for the model material. The TEC unit was mounted to the underside of the lid of the fabricated box as shown in Figure 26. Due to the nature of the aluminum tray provided in the existing television set, the new model to be tested did not initially fit into the original cabinet. As a result, three flow columns had to be fabricated so as to allow air flow to extend from the newly fabricated box to the LCD components. These rectangular columns (with the cross-sectional dimensions of the holes shown in Figure 24) were constructed from a .25 inch thick cardboard material and taped from the fabricated box to the bottom of the existing aluminum tray allowing air flow to reach the LCD's.

Testing Procedure and Results

TEST 1: TWO CENTRIFUGAL FANS AND TEC

The testing procedure for the final design configuration is similar to that described in the fan and TEC sections of this report. Once the centrifugal fans and the thermoelectric cooler were in place as described in the experimental setup, the leads were connected to two power sources used to run each component. Both fans were run in the upper end range for voltage (10 to 14 volts) and the TEC performed at a voltage of 10 volts. Before turning the power supplies on, the initial temperatures of the LCD components were measured and recorded as shown in Table 12. The results gained throughout this test were recorded in a 2-hour period in order to allow the temperature of the LCDs to reach steady-state values.

LCD	t = 0	t = 20 min	t = 40 min	t = 60 min	t = 80 in	t = 100 min	t = 120 min	Delta T
RED	21.8	51.0	56.8	57.6	57.4	57.2	57.0	35.2
GREEN	21.8	51.2	48.2	47.6	46.4	46.6	46.6	24.8
BLUE	21.8	49.0	47.8	51.0	50.0	50.2	51.2	29.4

Table 12. Average temperatures for the final design configuration with fans and TEC. All temperatures are expressed in °C.

These values can be readily compared to the experimental results shown in Table 2 of the baseline tests run with the original axial fan arrangement. Figure 27 is a plot of time versus temperature for the results obtained in this test.

TEST 2: CENTRIFUGAL FANS ONLY

In the second test run on the final design configuration, the performance of the centrifugal fans was desired without the use of the thermoelectric cooler. The purpose of this test was to examine the benefits associated with the use of the TEC. The experimental procedure for this test is identical as that described in TEST 1 except for the use of the thermoelectric cooler. Run at approximately 12 volts, the fans produced the following results as shown in Table 13.

LCD	t = 0	t = 20 min	t = 40 min	t = 60 min	t = 80 in	t = 100 min	t = 120 min	Delta T
RED	16.4	38.8	43.4	46.6	42.0	44.8	45.3	28.9
GREEN	16.4	36.8	40.4	44.2	39.8	43.2	44.0	27.6
BLUE	17.1	42.4	44.0	47.0	42.2	46.0	47.1	30.0

Table 13. Average temperatures for the final design configuration with two centrifugal fans only. All temperatures are expressed in °C.

Figure 28 shows the experimental results in the form of a plot of time versus temperature for the dual centrifugal fan setup.

Fan Noise Considerations

For nearly all electronics applications in which fans are utilized, the issues of fan noise and acceptable decibel ranges are of primary importance. For the tests run on the existing forced convection system, centrifugal fans produced higher decibel output when compared to their axial counterparts. In the published report from Philips, a decibel range for the axial fans range from 28-32 dB, whereas, Comair-Rotron's centrifugal fans taken from the two hair dryers list a minimal decibel rating of 53 dB. One recommendation for utilizing the centrifugal fans while addressing the fan noise issue would be to run the centrifugal fans at a lower shaft speed. This alternative, however, would decrease the volumetric flow rate and possibly hinder the effectiveness of the fans. Another suggestion for decreasing the decibel output would be to line the interior of the fan housing and the flow box with a noise absorption material such as cork, rubber, or plastic foam. Applied in such a way, the absorptive material would, in effect, dampen the unwanted noise produced by the centrifugal fans. As shown in the various testing results in this section of the report and other testing results of interest located in Appendix C, dual centrifugal fan usage produces better results for optimizing forced air convection through the specific channel placement in the testing unit.

Comparison to Theoretical Study

Sample calculations were made to analyze the potential for the design configuration to provide promising results in cooling the three LCDs located in the Philips projection television. The sample calculations were used to determine the amount of heat that could be dissipated from each of the LCDs using the test configuration made up of the TEC and 2 centrifugal fans. The following table is a comparison of the watts dissipated from each of the three LCDs for the Philips configuration and our final design configuration.

LCD	Philips Configuration (Axial Fans)	Final Design Configuration (TEC / 2 Centrifugal Fans)	Percent Increase
RED	0.62 watts	1.96 watts	216.0%
GREEN	1.24 watts	1.89 watts	52.4%
BLUE	0.96 watts	2.54 watts	164.8 %

Table 14: Comparison of the Watts dissipated for each LCD between Philips configuration and Final Design Configuration

This comparison provides a quantitative analysis of the theoretical performance of the new cooling configuration and shows the improvement in the condition of the LCD components. Although the experimental results do not correlate very well with the calculated results presented in Table 14, we feel that the proposed cooling system would be effective if the fabricated parts were of professional quality.

Cost Analysis

The centrifugal fans mentioned in the testing portion of this report were obtained from a Conair hairdryer. According to the manufacturing department at the Conair plant, these same fans can be purchased for $0.10 each if purchased in 200,000 unit bulk. The cost associated with the DT12-8 TEC unit tested in this project is $26.00 if bought in units of 100-250. This price would increase $6.50 if only 1-9 units were purchased at a time. Consequently, the price per television unit for the proposed cooling system would be approximately $26.10.

Figure 23. Experimental Setup of Final Design Configuration

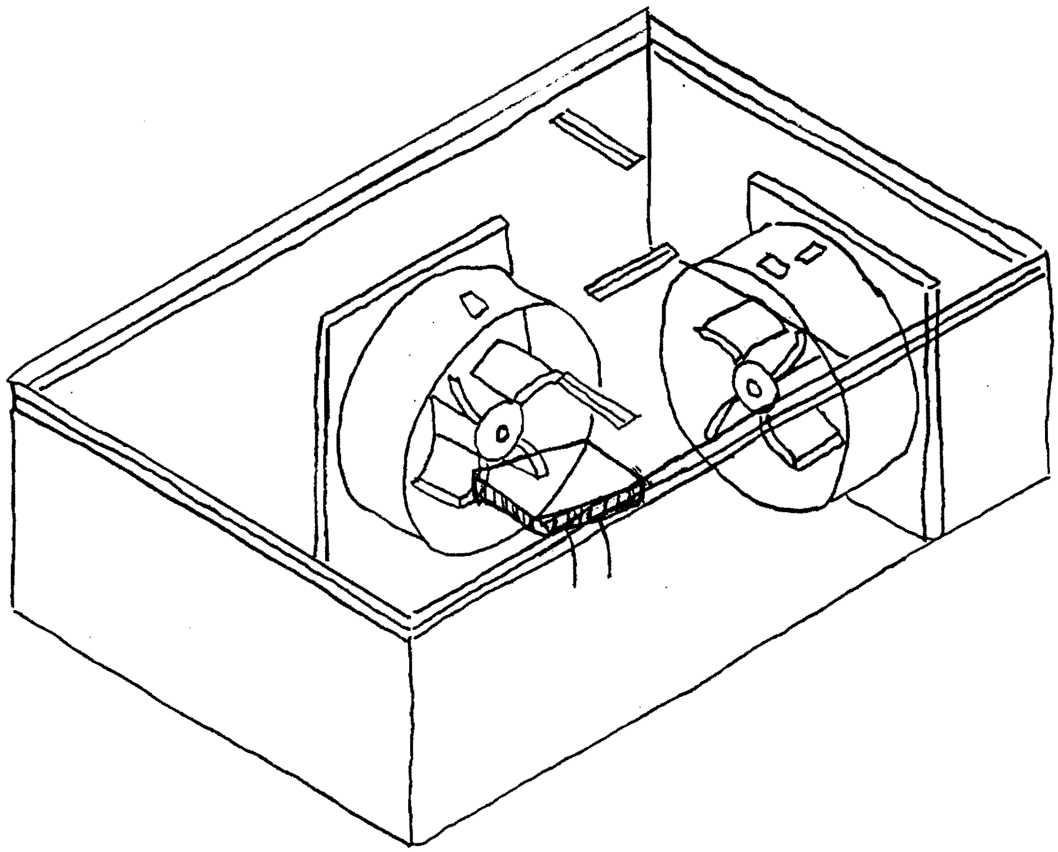


Figure 24. Placement of Flow Channels on Top of Sheet Metal Box

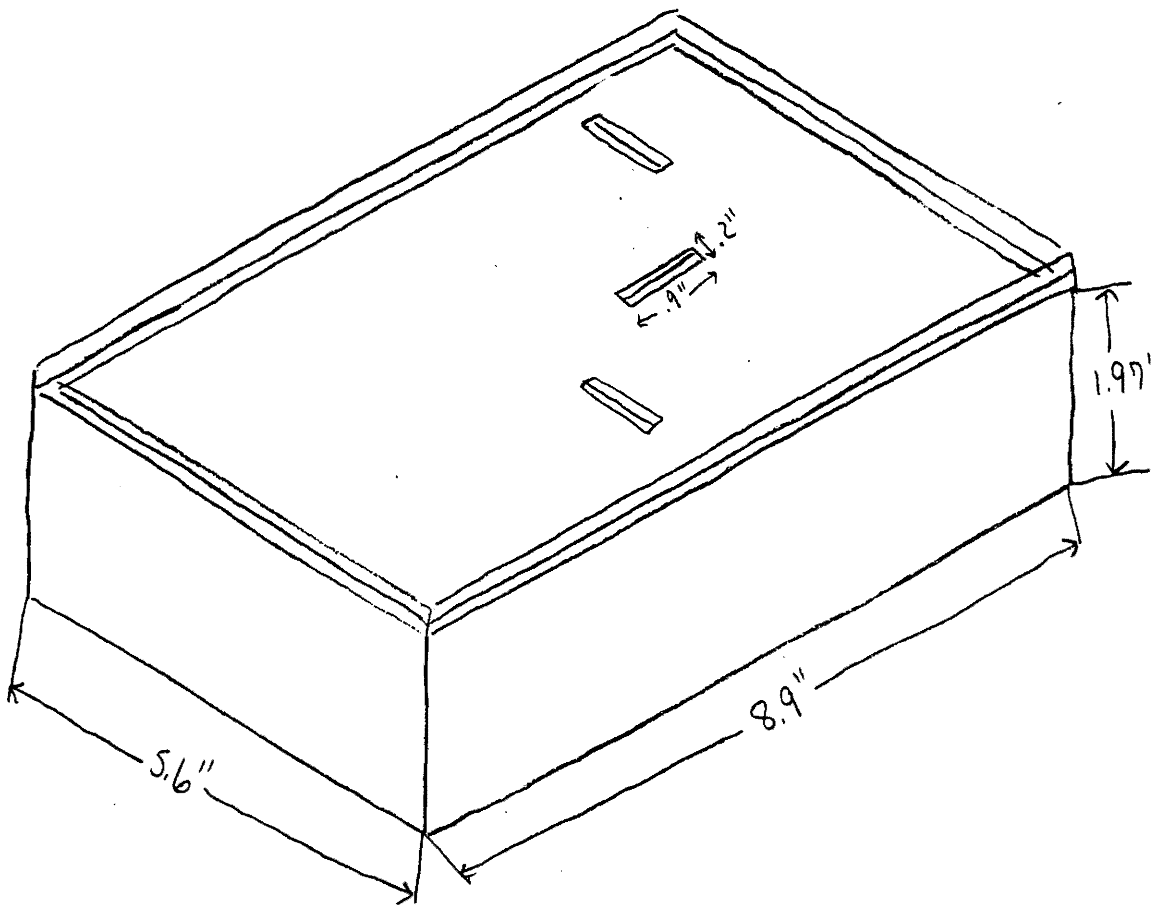
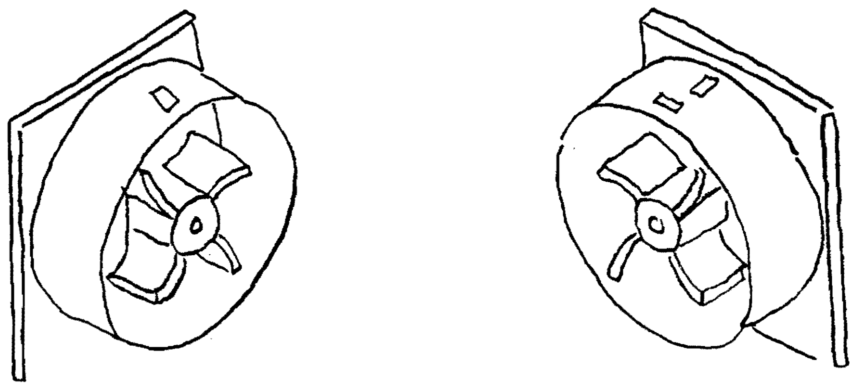


Figure 25. Centrifugal Fan Housings Used in Experimental Testing



Cylindrical Casing Diameter = 2.17"

Blade Width = 0.60"

Blade Length = 1.38"

Figure 26. Thermoelectric Cooler Placement

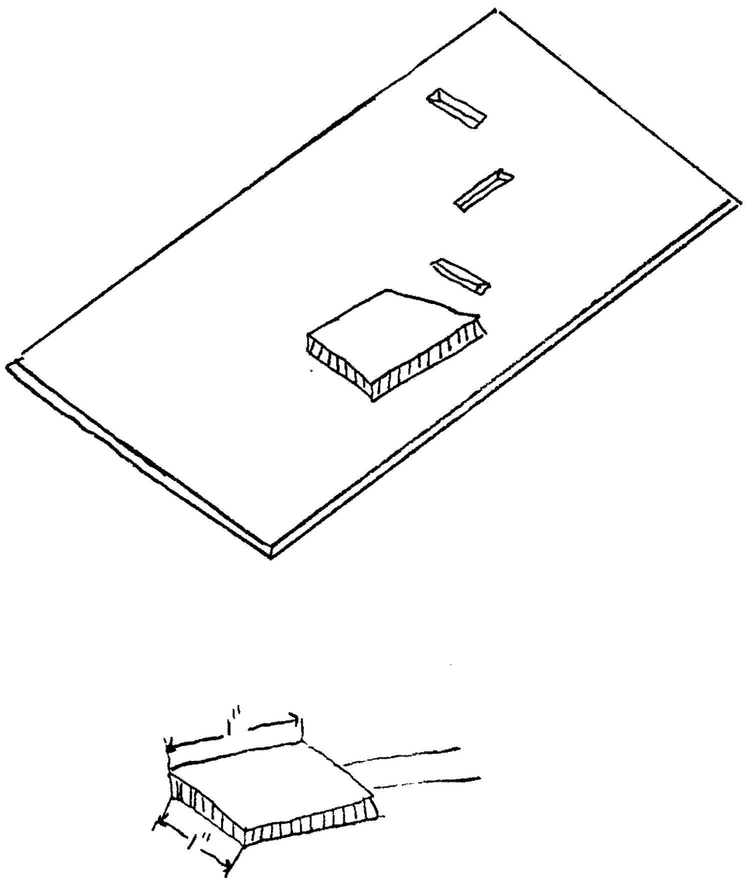


Figure 27. Performance Plot of TEC and Two Centrifugal Fans

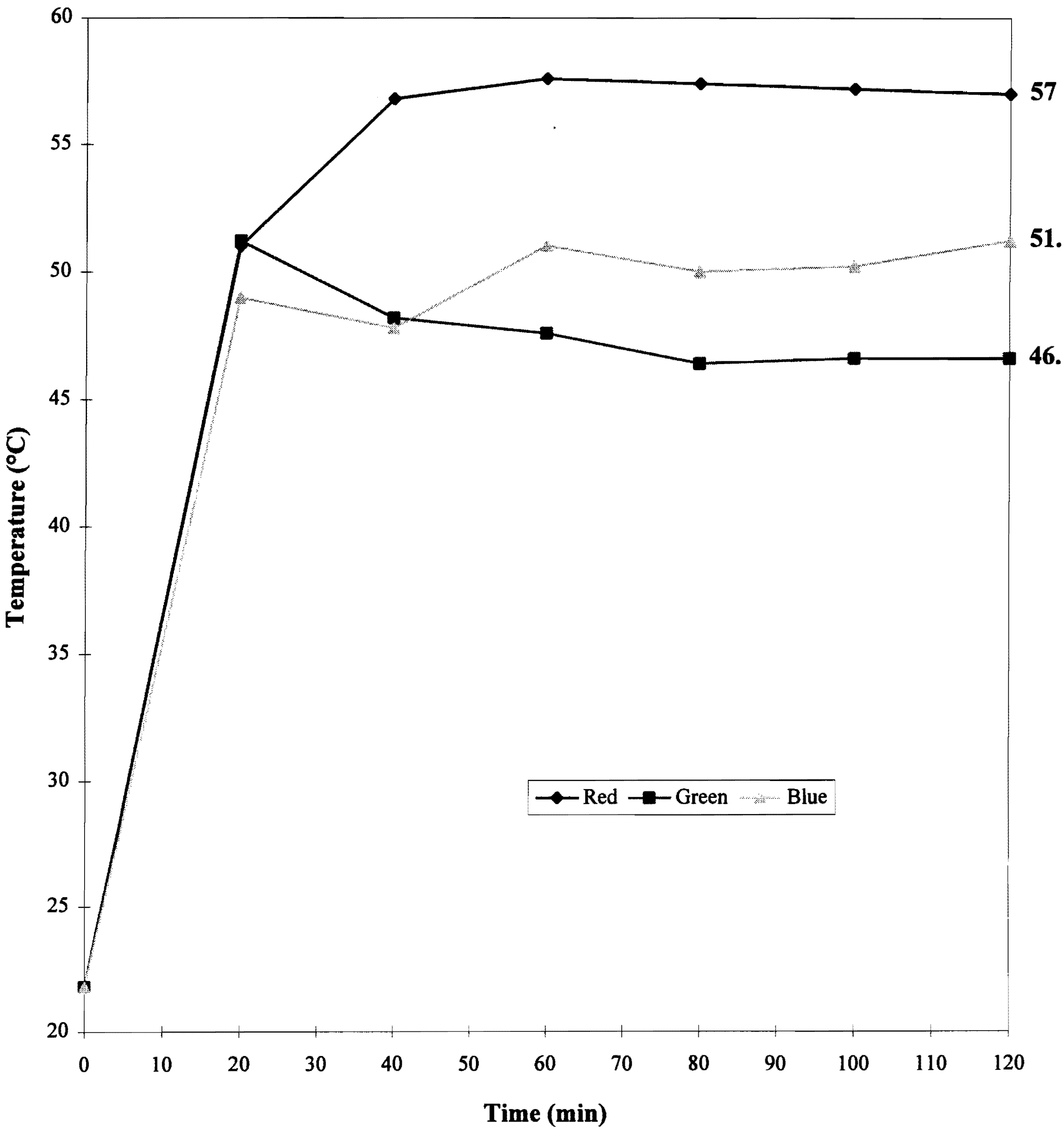
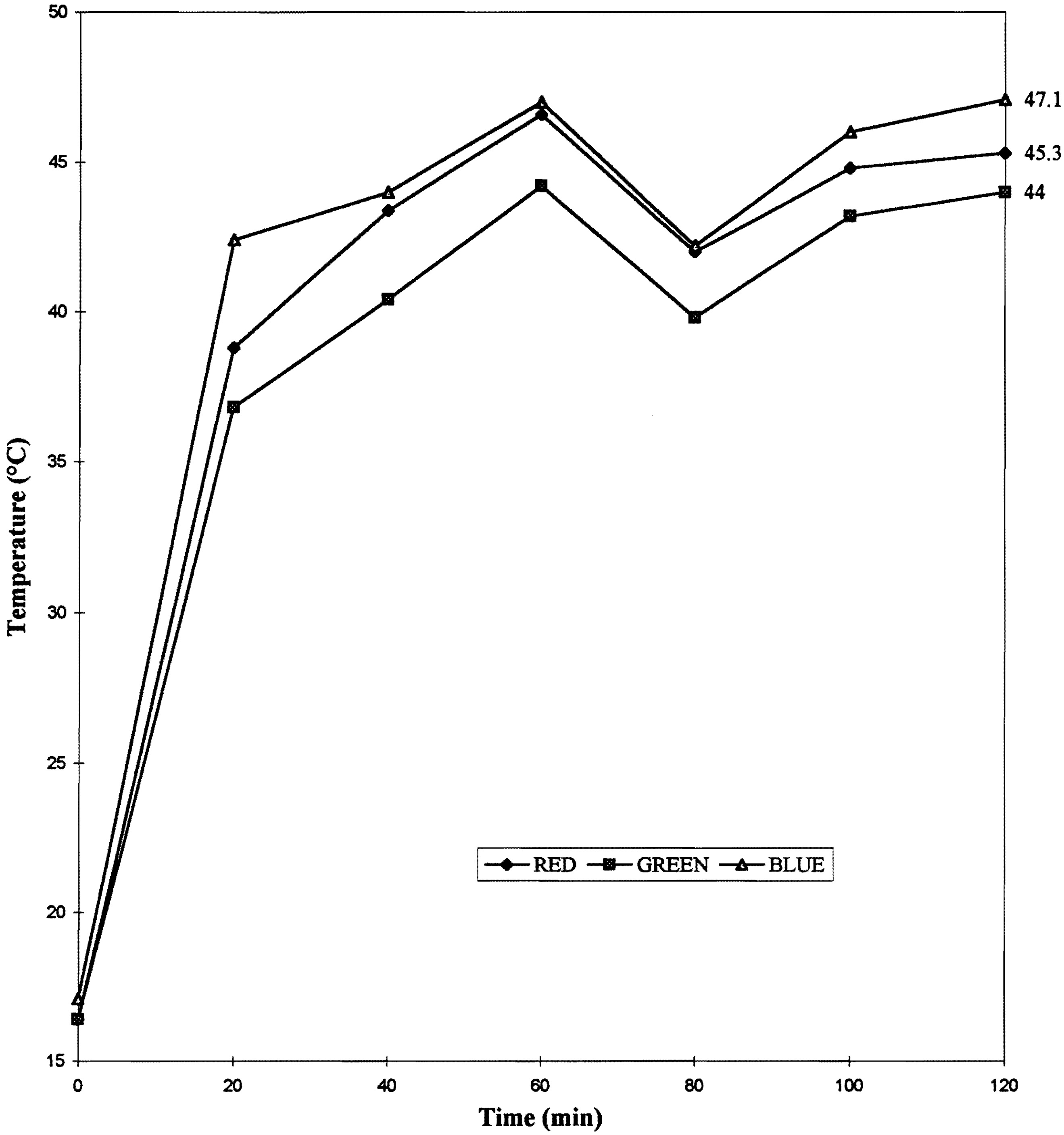


Figure 28. Performance Plot of Two Centrifugal Fans



CONCLUDING RECOMMENDATIONS

Based on the tests run on the final configuration developed throughout the duration of the semester, it is the recommendation of Team #1 that the dual-centrifugal fan arrangement when used with a thermoelectric cooling device is the best solution for the current heating problem. A sound-proofing material should be used as described in the Final Design portion of the report in order to minimize the decibel level associated with the operating centrifugal fans. Although sheet metal was used for the entire model in the testing procedure, only the top portion of the box needs to be constructed of a material with high thermal conductivity. Using a plastic material, as was used in the original television set, for the rest of the fan housing may reduce manufacturing costs and consequently be more desirable. If this option is explored, a material such as aluminum should be used for the top of the flow box to maximize the heat transfer from the thermoelectric cooling device.

APPENDICES TABLE OF CONTENTS

APPENDIX A

HEAT PIPES

A heat pipe is a hollow tube type of enclosed structure containing a fluid that transfers enormous amounts of heat when it evaporates. Once it condenses, the fluid is then brought back to its starting point by a wick. As shown in Figure A.1, the heat pipe can be divided into three sections: an evaporator section, a transport section, and a condenser section.

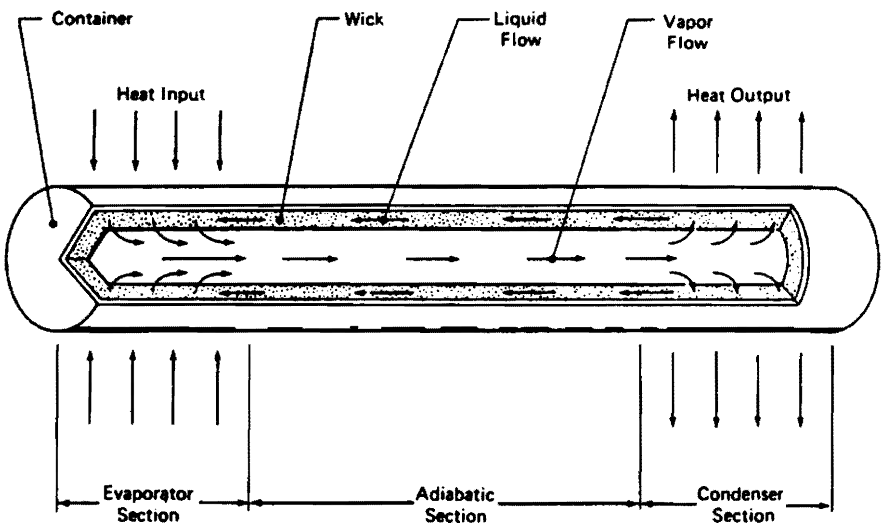


Figure A.1. Schematic diagram of the three stages of a heat pipe.

The evaporator section is normally placed near a direct heat source. At this end of the tube, heat enters and is absorbed by the fluid enclosed by the tube. The fluid then boils or vaporizes and a force is created which moves the fluid into the transport section and ultimately into the condenser section. The heat exits out of the condenser end forcing the vapor to condense back to liquid form. The liquid is then returned to the evaporator section by the capillary wick.

Heat pipes are used primarily to transport heat from the source to the sink without any external power. They are also effective in eliminating hot spots and can accept heat that has a high power density. Most heat pipes are positioned external to the device they are cooling and can readily be used in conjunction with other methods of cooling. In some applications, cooling fins are added to the condenser end of a heat pipe to enhance natural and forced convection cooling. Heat pipes can also be bent to reach heat concentrations and hot spots in remote areas of a chassis.

The advantages of the use of heat pipe technology are similar to that of liquid heat sink applications. Heat pipes operate without outside power sources and have no moving parts which cause unwanted noise. Also, heat pipes can be used in any type of orientation and can transfer heat hundreds of times better than any solid metal conductor.

THERMOSYPHONS

In a typical application, the thermosyphon is simply a tube enclosing a fluid which circulates under the influence of thermally induced buoyancy forces. Thermosyphons are one of several ways of trying to solve the problem of heat dissipation within electronic devices. Figure A.2 shows the transient response of the thermosyphon.

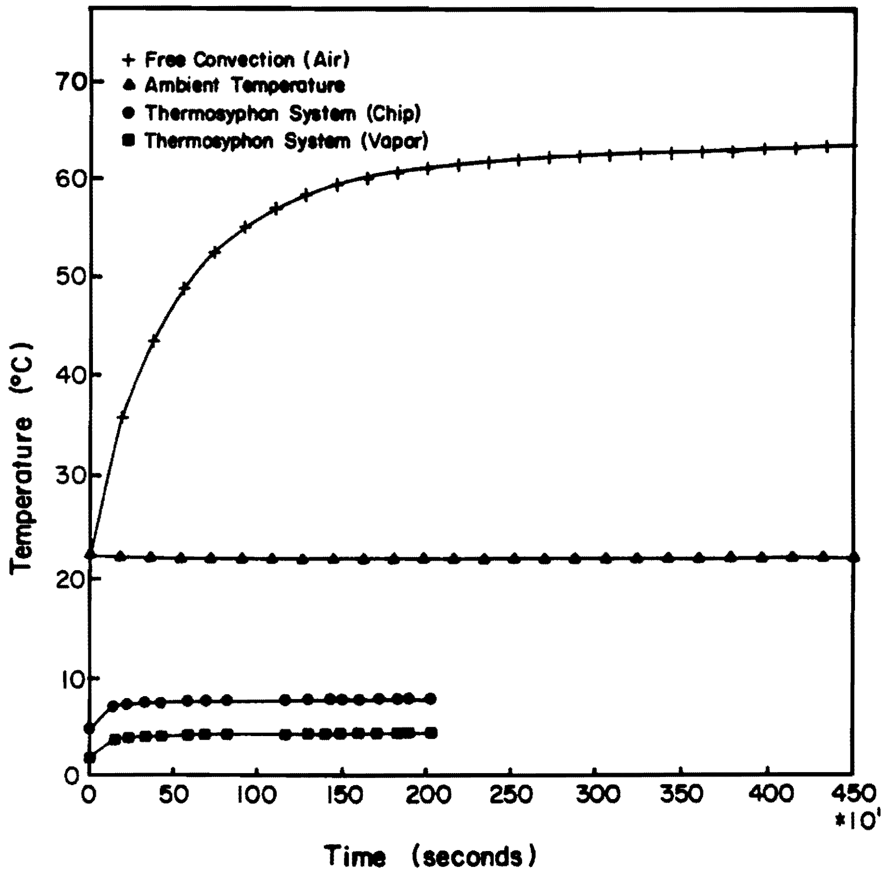


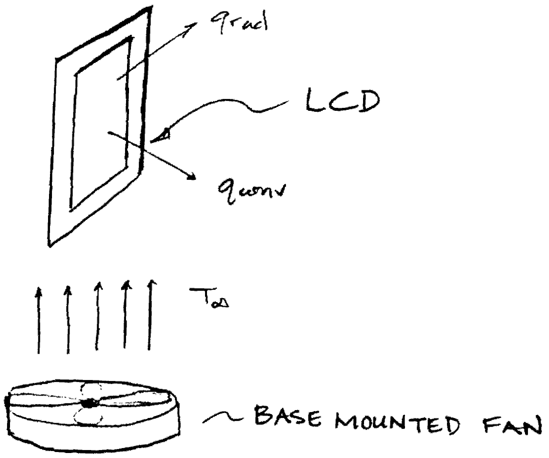
Figure A.2. Transient performance of thermosyphon.

A thermosyphon system consists of an evaporator, condenser, and a vapor and liquid passage. Electronic components such as integrated circuit (IC) chips are immersed in a pool of refrigerant and placed in the evaporator of the thermosyphon system. This is a form of pool boiling. When electrical energy is generated in the IC chips, the refrigerant begins to boil and vaporize inside the evaporator. The boiling liquid creates a vapor pressure that forces the vapor through the vapor passage to the condenser. The vapor is then condensed back to liquid form and returns to the evaporator through the liquid passage. The advantage of this system setup is that the fluid will always be at saturated conditions and vaporization can occur at any heat flux.

The pressure difference between the evaporator and condenser drives the vapor mass transfer, while the liquid flow is driven by the hydrostatic pressure resulting from the elevation difference of the condenser and pool height in the evaporator. Problems may arise in the mass transfer if the flow velocity is limited by friction within the system. For prolonged operation, the mass flow rates of the vapor and liquid must be equal.

APPENDIX B

FORCED CONVECTION ANALYSIS



$$\boxed{q_x = q_{convection} + q_{radiation}}$$

$h_{air} = 35\ W/m^2 \cdot K$

$A = 2(1.3\ in)^2$

$= 2.18 \times 10^{-3}\ m^2$

$T_{s\,max} = 338\ K$

$T_{\infty\,max} = 313\ °K$

$$q_{convection} = h_{air} A (T_s - T_\infty)$$

$$= (35\ W/m^2 K)(2.18 \times 10^{-3}\ m^2)(25\ °K)$$

$$= 1.91\ \text{Watts}$$

$$q_{radiation} = \sigma \epsilon A (T_s + T_{sur})(T_s^2 + T_{sur}^2)(T_s - T_{sur})$$

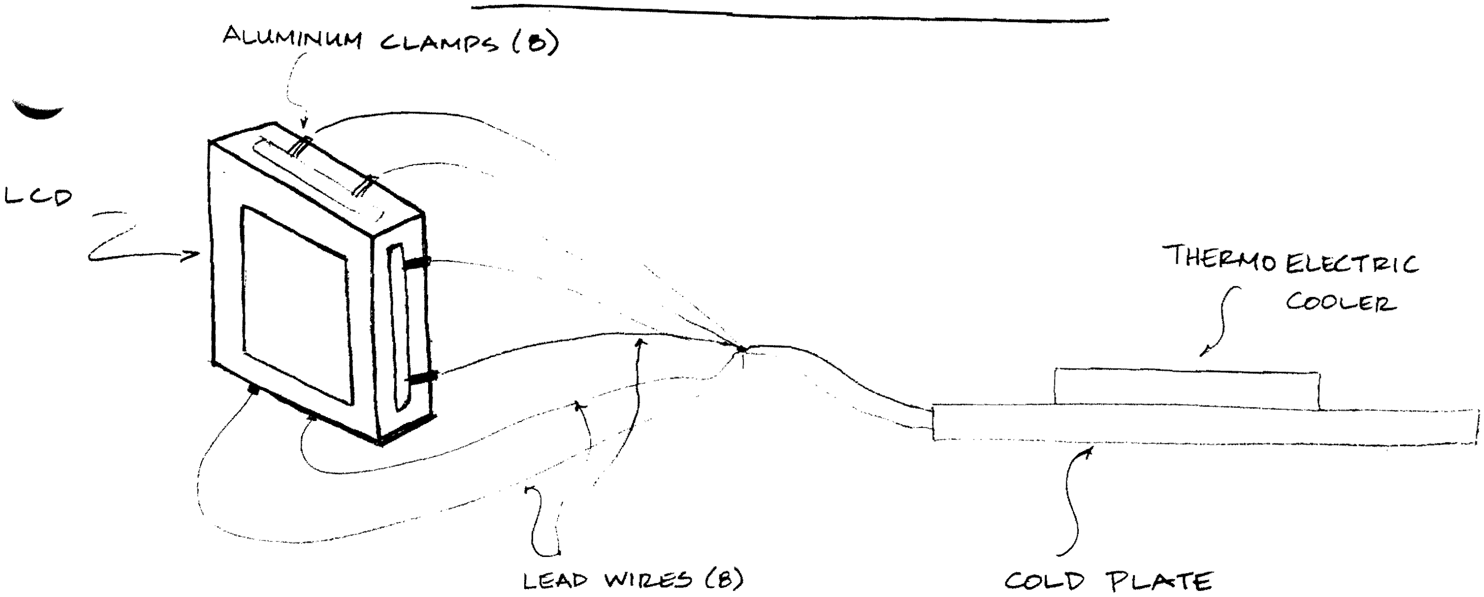
$$= (5.67 \times 10^{-8})(1)(2.18 \times 10^{-3}\ m^2)(338 + 313)(338^2 + 313^2)(25)$$

$$= 0.427\ \text{Watts}$$

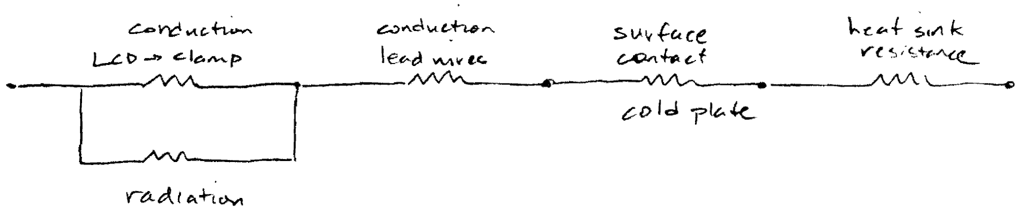
$$q_{total} = 1.91\ \text{Watts} + 0.427\ \text{Watts}$$

$$= \underline{\underline{2.34\ \text{Watts}}}$$

THERMO-ELECTRIC COOLER



THERMAL RESISTANCE MODEL



ASSUMPTIONS: CLAMPS are 2 mm thick Aluminum

Area = $1.135 \times 10^{-3}\ m^2$

Lead Wires are Copper : $L = 6\ cm$, $r = 2\ mm$, $A = 1.256 \times 10^{-5}\ m^2$

① $R_{radiation} = \dfrac{1}{h_r A} = 58.56\ °K/W$

② $R_{conduction} = \dfrac{L}{kA} = \dfrac{0.002\ m}{(237\ W/mK)(1.135 \times 10^{-3} m^2)} = 0.007\ °K/W$

③ $R_{conduction}$ (lead wires) $= \dfrac{L}{kA} = \dfrac{0.06\ m}{(401\ W/mK)(1.256 \times 10^{-5}\ m)} = 12.0\ K/W$

④ $R_{\text{conduction through cold plate}} = \frac{L}{KA} = \frac{0.002\ m}{(237\ W/mK)(1.135 \times 10^{-3}\ m^2)} = 0.007\ °K/W$

⑤ Heat Sink Resistance $= \frac{T_1 - T_2}{Q}$

$$= \frac{(50-40)°C}{2.34\ W}$$

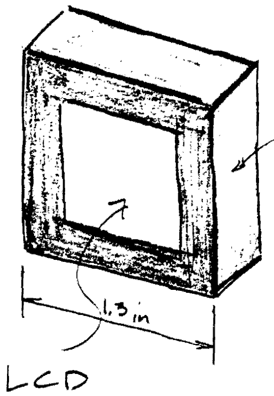
$$= 4.27\ °C/W$$

$$R_{total} = \left(\frac{1}{\frac{1}{58.56} + \frac{1}{8(0.007)}} \right) + 1.5\ °K/W + 8(0.007\ °K/W) + 4.27\ °K/W$$

$$R_{TOT} = 5.88\ K/W$$

$$q = \frac{\Delta T}{R_{TOT}} = \frac{25\ K}{5.88\ °K/W} = \underline{\underline{4.25\ \text{Watts}}}$$

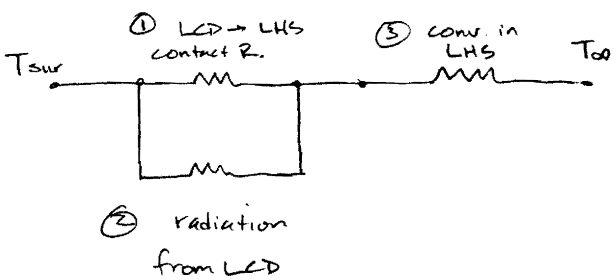
LIQUID HEAT SINK ANALYSIS



LIQUID HEAT SINK (shaded areas)

APPLICATION: L.H.S is draped over metal frame of LCD.

THERMAL RESISTANCE MODEL



Area of LHS $= 0.88\,in^2\,(2) = 1.76\,in^2$

$= 1.135 \times 10^{-3}\ m^2$

$h_r = \sigma\epsilon\,(T_s + T_{sur})(T_s^2 + T_{sur}^2)$

$= (5.67 \times 10^{-8})(1)(338 + 313)(338^2 + 313^2)$

$= 7.833\ W/m^2K$

① $R_{contact} = 2\ °K/W$ (source: Aavid technical report)

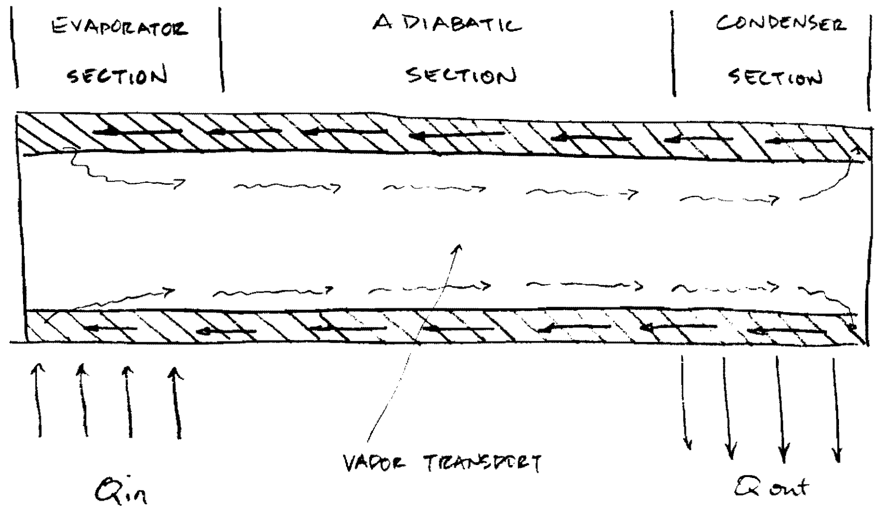
② $$R_{radiation} = \frac{1}{h_r A} = \frac{1}{(7.833\ W/m^2K)(2.18 \times 10^{-3}\ m^2)} = 58.56\ °K/W$$

③ $$R_{conv.\ in\ LHS} = \frac{1}{hA} = \frac{1}{(0.15\ W/m^2K)(1.135 \times 10^{-3}\ m^2)} = 0.587\ °K/W$$

$$R_{tot} = \frac{1}{\left(\frac{1}{58.56} + \frac{1}{2}\right)} + 0.587\ °K/W = 2.52\ °K/W$$

$$q = \frac{\Delta T}{R_{tot}} = \frac{25\ °K}{2.52\ °K/W} = \underline{9.92\ \text{Watts}}$$

HEAT PIPE CALCULATIONS



DIMENSIONS:

Wall thickness (t_w) = 1.0 mm

Outer Diameter (D_o) = 20 mm

Inner Diameter (D_i) = 18 mm

Core Diameter (D_v) = 16 mm

Evaporator Length (L_e) = 100 mm

Condenser Length (L_c) = 80 mm

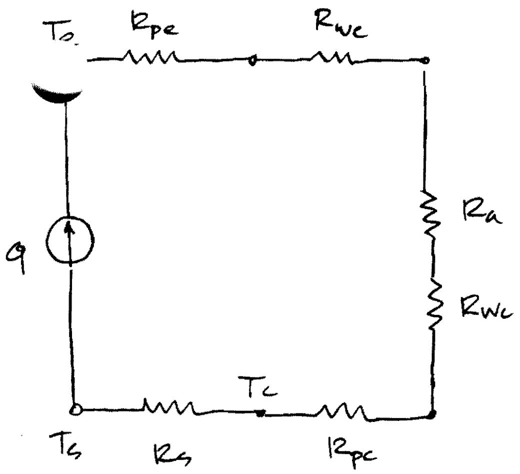
Effective Conductivity of Wick = 1.998 W/m·K

Thermal Conductivity of Copper Heat Pipe = 401 W/m·K

Temperature at Surface = 338 °K

Temperature of Surroundings = 313 °K

HEAT PIPE CALCULATIONS



Definition of Terms

$q \equiv$ heat in

$T_e \equiv$ temperature at surface of evaporator

$R_{pe} \equiv$ resistance of pipe wall at evaporator.

$R_{we} \equiv$ resistance of the wick at the evaporator

$R_a \equiv$ resistance of the adiabatic section

$R_{wc} \equiv$ resistance of the wick at the condenser

$R_{pc} \equiv$ resistance of the pipe at the condenser

$T_c \equiv$ temperature at the surface of the condenser.

$R_s \equiv$ resistance between condenser + environment

$T_s \equiv$ temperature at the surface

RESISTANCE CALCULATIONS:

$$R_{pe} = \frac{\ln(d_o/d_i)}{2\pi L_e k_m} = \frac{\ln(20/18)}{2\pi(100\times10^{-3}\,m)(1.998\ W/m\cdot{}^\circ K)} = 4.2\times10^{-4}\ {}^\circ K/W$$

(2) $$R_{we} = \frac{\ln(d_i/d_v)}{2\pi L_e k_e} = \frac{\ln(18/16)}{2\pi(100\times10^{-3}\,m)(1.998\ W/m\cdot{}^\circ K)} = 9.38\times10^{-2}\ {}^\circ K/W$$

(3) $R_a = 0$ Because there is little vapor pressure loss in adiabatic section

(4) $$R_{wc} = \frac{\ln(d_i/d_v)}{2\pi L_c k_e} = \frac{\ln(18/16)}{2\pi(80\times10^{-3}\,m)(1.998\ W/m\cdot K)} = 0.1173\ {}^\circ K/W$$

(5) $$R_{pc} = \frac{\ln(d_o/d_i)}{2\pi L_c k_m} = \frac{\ln(20/18)}{2\pi(80\times10^{-3}\,m)(401\ W/m\cdot K)} = 5.23\times10^{-4}\ {}^\circ K/W$$

(6) $$R_s = \frac{1}{hA} = \frac{1}{(25\ W/m^2\text{-}K)(0.0258064\ m^2)} = 1.55\ °K/W$$

$$R_3 = R_{pe} + R_{we} + R_a + R_{wc} + R_{pc} + R_s$$

$$= (4.2\times10^{-4} + 94\times10^{-4} + 0 + 0.1173 + 5.23\times10^{-4} + 0.5741)\ °K/W$$

$$= 0.6933\ °K/W$$

$$\frac{1}{Re} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3}$$

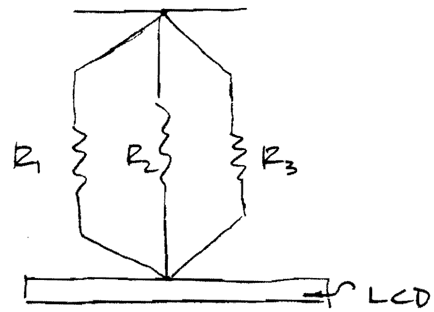
$$= \frac{1}{62.67} + \frac{1}{58.56} + \frac{1}{1.55}$$

$$= 0.6781$$

$$\therefore\ Re = 1.475$$

$$q = \frac{\Delta T}{Re} = \frac{25\ °C}{1.475\ °C/W} = 16.95\ \text{Watts.}$$

HEAT PIPE ANALYSIS



$R_1 \equiv$ convection

$$h = \frac{\bar{N}u_L \, k}{0.03302} = 7.319 \ W/m^2 \cdot {}^\circ K$$

$$A = 0.00218 \ m^2$$

$$R_1 = \frac{1}{hA} = \frac{1}{(7.319 \frac{W}{m^2 K})(0.00218 m^2)} = 62.67 \ {}^\circ K/W$$

$R_2 \equiv$ radiation

$$hr = \epsilon\sigma (T_s + T_{sur})(T_s^2 + T_{sur}^2)$$

$$= 1(5.67 \times 10^{-8})(338 + 313)(313^2 + 338^2)$$

$$= 7.833$$

$$R_2 = \frac{1}{hr\,A} = \frac{1}{(7.833 \frac{W}{m^2 K})(0.00218 m^2)} = 58.56 \ {}^\circ K/W$$

THERMOSYPHON ANALYSIS

DESCRIPTION:

6 x 8 Vertical Array of IC Chips

IC chips 12 mm x 20 mm

Total Area of Array = 128 cm^2

Area of Condenser = 0.0145 m^2

Area of Evaporator = 0.0102 m^2

Maximum Chip Surface Temperature = 57°C

Temperature of System = 20°C

Total Power Dissipated = 12 Watts

HEAT TRANSFER STRIP ANALYSIS

NOTE: Heat Dissipation varies with size and placement of the conductive heat transfer strip. Individual testing is advised to evaluate the maximum effectiveness of the strip.

FAN NOISE ANALYSIS

FAN INDUCED FLOW RATE

$$\dot{Q} = \frac{fW}{T_{OUT} - T_{IN}}$$

$f = 3.1$

W = TOTAL POWER [illegible]

T_{OUT} = MAXIMUM CABINET TEMPERATURE BEFORE FAILURE; 65°C

T_{IN} = AMBIENT AIR TEMP; 40°C

$$\dot{Q} = \frac{3.1(132)}{25} = 16.3[illegible]3 \text{ cfm} = 7.72 \text{ l/s}$$

USING FIGURE [illegible] FROM THE [illegible] [illegible]

[illegible]

[illegible]

[illegible]

[illegible]

[illegible]

L_T = NOISE TOTAL [illegible]

L_i = INDIVIDUAL NOISE SOURCE PRESSURE LEVEL

PLACING [illegible] SIDE

$$L_T = 10 \log\left(10^{[illegible]} + 10^{29/10}\right)$$

$= \underline{35.0 \text{ dB}}$

MAXIMUM PRESSURE OR AIRFLOW MAY BE INCREASED WHILE KEEPING THE OTHER PARAMETER NEAR CONSTANT

- WOULD FAVOR PARALLEL

APPENDIX C

Appendix C. Flow Rate Results from Fan Configuration Testing

BASELINE MEASURES

* Use of Mini-Annomometer Series 490

	TEST 1	TEST 2	TEST 3	
RED	550	550	550	FT/MIN
GREEN	550	550	540	FT/MIN
BLUE	800	800	850	FT/MIN

* Test involved all components removed from unit except the lamp. The device was placed in the center of each slot in the base. 2 minutes were allotted for unit warm up and 1 minute for calcualtions.

* Use of Cole-Parmer Tri-Sense Annomometer

	TEST 1	TEST 2	TEST 3	
RED	660	664	648	FT/MIN
GREEN	462	407	519	FT/MIN
BLUE	928	950	956	FT/MIN

CENTRIFUGAL FAN TESTING

* Placed in built apparatus; Voltage used – 9V
* Exposed hole cut to draw in fresh air

	TEST 1	TEST 2	TEST 3	
RED	1000	1000	1000	FT/MIN
GREEN	700	1000	900	FT/MIN
BLUE	1250	1100	1300	FT/MIN

80mm AXIAL FAN AND CENTRIFUGAL FAN TESTING

* Plastic sleeve is placed on top of 80mm fan

	TEST 1	TEST 2	TEST 3	
RED	300/450	400/550	350/500	FT/MIN
GREEN	200/550	200/600	200/600	FT/MIN
BLUE	2500/3200	2900/3200	2900/3200	FT/MIN

Appendix C. Flow Rate Results from Fan Configuration Testing

DUAL CENTRIFUGAL FAN PLACEMENT

	TEST 1	TEST 2	TEST 3	
RED	500/1200	900/2200	750/2000	FT/MIN
GREEN	750/1500	1500/2000	1000/2200	FT/MIN
BLUE	500/1200	400/2500	400/2400	FT/MIN

* Alternate cup was placed over blue channel. The channels for red and blue were switched, but the fan arrangement remained the same.

	TEST 1	TEST 2	TEST 3	
RED	500/1500	400/1400	450/2000	FT/MIN
GREEN	1000/1500	700/1000	750/1500	FT/MIN
BLUE	2700/3250	2700/3250	2700/3100	FT/MIN

DUAL 80mm AXIAL FAN TESTING

* Plastic sleeve is placed on top of 80mm fan

* Ran at 14V

	TEST 1	TEST 2	TEST 3	
RED	350/400	350/500	400/550	FT/MIN
GREEN	400/900	200/1100	200/1100	FT/MIN
BLUE	50/900	100/1100	50/1100	FT/MIN

120mm AXIAL FAN TESTING

* Plastic sleeve is placed on top of 120mm fan

	TEST 1	TEST 2	TEST 3	
RED	100/200	200/300	300/400	FT/MIN
GREEN	200/300	200/400	250/450	FT/MIN
BLUE	500/1500	750/1500	500/1000	FT/MIN

APPENDIX D

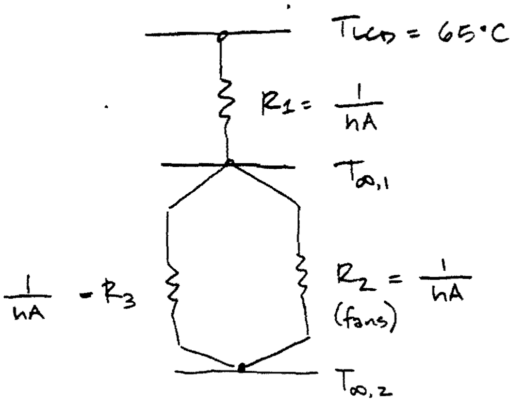
Appendix D. Experimental Setup Equipment List

1. **Two Centrifugal Fans**
2. **Two DC Power Sources**
3. **Duct Tape**
4. **Mini-Anemometer**
5. **Paper Cups**
6. **Sheet Metal Box**
7. **Soldering Iron**
8. **Temperature Display Box**
9. **Thermocouple Control Panel**
10. **Type T Thermocouples**

APPENDIX E

Appendix E. Calculations Associated with Final Design Configuration

(A) Thermal Resistance Circuit :



$$\text{Heat Transfer } (q) = \frac{T_{LED} - T_{\infty,2}}{R_1 + \frac{1}{\frac{1}{R_3} + \frac{1}{R_4}}}$$

(B) Methodology For Convection Calculations

(1) Flow Geometry - Flow is over a flat surface plate across an enclosed channel.

(2) Reference Temperature and Pertinent Fluid properties At The Reference Temperature

from Table A4: fluid : Air ~ Assumed cooled TEC air at $T = 60°F = 15.6°C = 288K$

Interpolated data : $\{ \rho = 1.22$ kg/m³ $\quad c_p = 1.00676 \frac{kJ}{kg\,K} \quad \mu \cdot 10^7 = 178.6 \frac{N \cdot s}{m^2}$

$\nu \cdot 10^6 = 14.82$ m²/s $\quad k \cdot 10^3 = 25.34 \frac{W}{m \cdot K} \quad \alpha \cdot 10^6 = 20.916$

$Pr = 0.71012 \}$

(3) Reynolds # Calculations (defines the ratio of inertia & viscous forces)

$$Re_{x,c} = \frac{\rho u_\infty L}{\mu} = 5 \times 10^5$$

SAMPLE CALCULATIONS

(C) Reynolds + Nusselt Number Calculations :

$$\underset{(red)}{Re} = \frac{1.22\ kg/m^3\,(6.14\ m/s)(0.03302\ m)}{178.6\times10^{-7}\ \frac{N\text{-}s}{m^2}} = 13622 < 5\times10^5$$

$\therefore$ laminar

$$\underset{(green)}{Re} = \frac{1.22\ kg/m^3\,(5.46\ m/s)(0.03302\ m)}{178.6\times10^{-7}\ \frac{N\text{-}s}{m^2}} = 12315\text{ , laminar}$$

$$\underset{(blue)}{Re} = \frac{1.22\ kg/m^3\,(14.16\ m/s)(0.03302\ m)}{178.6\times10^{-7}\ \frac{N\text{-}s}{m^2}} = 31939\text{ , laminar}$$

$$\underset{(red)}{Nu} = \frac{hL}{K} = 0.664\,Re_x^{1/2}\,Pr^{1/3} = 0.664\,(13622)^{1/2}(0.71)^{1/3} = 69.1$$

$$h = \frac{69.1\,k}{L} = \frac{69.1\,(26.3\times10^{-3}\ W/m\cdot K)}{0.03302\ m} = 55.03\ W/m^2K$$

$$\underset{(green)}{Nu} = \frac{hL}{k} = 0.664\,Re_x^{1/2}\,Pr^{1/3} = 0.664\,(12315)^{1/2}(0.71)^{1/3} = 65.7$$

$$h = \frac{65.7\,K}{L} = \frac{65.7\,(26.3\times10^{-3}\ W/m\cdot K)}{0.03302\ m} = 52.4\ W/m^2\cdot K$$

$$\underset{(blue)}{Nu} = \frac{hL}{K} = 0.664\,Re_L^{1/2}\,Pr^{1/3} = 0.664(31939)^{1/2}(0.71)^{1/3} = 105.9$$

$$h = \frac{105.9\,(26.3\times10^{-3}\ W/m\cdot K)}{0.03302\ m} = 84.32\ W/m^2\cdot K$$

SAMPLE CALCULATIONS

(D) Heat Transfer Calculations

(1) Red LCD: (heat transfer calculations)

$$R_1 = \frac{1}{(55.05)(0.00218m^2)} = 8.33\ °K/W$$

$$R_2 = \frac{1}{hA} = \frac{1}{(21.7)(0.0016m^2)} = 28.8\ °K/W$$

$$R_3 = \frac{1}{hA} = \frac{1}{(88)(0.00218m^2)} = 5.21\ °K/W$$

$$R_{TOTAL} = 8.33 + \frac{1}{\frac{1}{28.8} + \frac{1}{5.21}} = 8.33 + 4.41 = 12.74$$

$$Q_{red} = \frac{\Delta T}{R_{TOTAL}} = \frac{65-40}{12.74} = \underline{\underline{1.96\ \text{Watts}}}$$

(2) Green LCD: (heat transfer calculations)

$$R_1 = \frac{1}{(52.4)(0.00218m^2)} = 8.75\ K/W$$

R_2 ; R_3 (same as above)

$$R_{TOTAL} = 13.16\ °K/W$$

$$Q = \underline{\underline{1.89\ \text{Watts}}}$$

(3) Blue LCD: (heat transfer calculations)

$$R_1 = \frac{1}{(84.32)(0.00218m^2)} = 5.44\ °K/W$$

$$R_{TOTAL} = 9.85\ °K/W$$

$$Q = \underline{\underline{2.54\ \text{Watts}}}$$